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THE FOURTH NATIONAL CONFERENCE ON ACOUSTICS VOL. I,C. ROOM AND ARCHITECTURAL ACOUSTICS

A. Stan, G. Silas et al.

Translation of "A IV-A Conferinta Nationala de Acustica. Vol. I,C. Acustica Arhitecturala Si a Salilor," Academy of the Socialist Republic of Romania, Bucharest, 29-31 May 1973, pp. 344-470

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16. Abstract

The papers presented deal with the optimization of the acoustical treatment not only of industrial plants but also of social-cultural buildings such as theaters and residences (apartment houses); in the latter connection, with due regard for differences in building materials, such as light prefabs, light concretes, etc.

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SECTION C *

ROOM AND ARCHITECTURAL ACOUSTICS

*For the prior sections, see the translation NASA TT F-15,375.

CONTENTS

SECTION C: ROOM AND ARCHITECTURAL ACOUSTICS

CONTRIBUTION TO THE ACOUSTIC STUDY OF THEATER AUDITORIUMS TRANSFORMABLE FOR DIFFERENT TYPES OF PRODUCTIONS
R. Belea, E. Harastaseanu

OPTIMIZING ACOUSTICAL TREATMENT
Nicolae Beuran, Septimiu Ramboiu, Ioan Farcas, Erich Halpert

NUMERICAL COMPUTATION OF ECHOGRAMS IN A ROOM: APPLICATION TO DETERMINATION OF INTELLIGIBILITY
Francois Santon

NEW THEORY ON THE REVERBERATION OF ROOMS
Jean Pujolle

ANECHOIC CHAMBER IN INDUSTRIAL PLANTS
E. Halpert, O. Juncu, R. Lorian, D. Marfievici, I. Mararu

ASPECTS OF CHECKING INSTALLATIONS FOR THE NOISE LEVEL OF ELECTRONIC EQUIPMENT
Petre Rotileanu, Mariana Augustin, Mircea Burtan

SOME ASPECTS OF THE BEHAVIOR OF MATERIALS USED IN MECHANICAL VIBRATION INSULATING TECHNIQUES
Constantin Pupazen, Cristian Ionid

EFFICIENT SYSTEMS OF SOUND-INSULATING FLOORS
Ioan Ion Giurgiu

CONTRIBUTIONS TO THE ACOUSTIC PROTECTION OF CIVIL BUILDINGS TAKING INTO ACCOUNT THE USE OF LIGHT PREFABS
Marius Smigelschi

CONCERNING THE DAMPING COEFFICIENTS OF EXTERIOR BUILDING ELEMENTS FOR AERIAL NOISE
V. Pocsa, M. Poppel, Al. Veres, L. Biborosch

TRANSPARENT SOUND SCREENS
Flaviu Bizo, Rozsa Draghici

CONCERNING THE SOUND INSULATION OF BUILDING ELEMENTS MADE UP OF LIGHT CONCRETES
Ioan Ion Giurgiu

OPTIMIZATION OF THE ACOUSTIC ABSORPTION COEFFICIENTS OF CERTAIN FUNCTIONAL ABSORBENTS
V. Pocsa, L. Biborosch, Al. Veres,
E. Halpert, R. Lorian, T. Botos

SOUND-ABSORBING SLABS AND STRUCTURES BASED ON GRANULAR MATERIALS (BOUND AND UNBOUND)
Serban Petre-Lazar, Gheorghe Popeea

COMPLEX SOUNDPROOFING OF INDUSTRIAL ROOMS
V. Pocsa, Al. Veres, L. Biborosch

HIGHLY EFFICIENT SPECIAL SOUND-ABSORBING SOLUTIONS
Mariana Ionescu, Serban Petre-Lazar

SOUND ISOLATION PROPERTIES OF A LIGHTWEIGHT DWELLING HOUSE

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THE FOURTH NATIONAL CONFERENCE ON ACOUSTICS
VOL. I,C.
ROOM AND ARCHITECTURAL ACOUSTICS

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CONTRIBUTION TO THE ACOUSTIC STUDY OF THEATER AUDITORIUMS TRANSFORMABLE FOR DIFFERENT TYPES OF PRODUCTIONS

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1. Introduction

The auditorium of the Bucharest National Theater has been equipped with a fixed seating section C and three mobile seating sections: A and B equipped with delimiting walls and D without a delimiting wall; sliding walls and descending floors. /345

By maneuvering the mobile seating sections and the other mobile elements: sliding walls and descending floors, the auditorium can be transformed into three functional types of room for different types of productions, namely:

a) An "Italian" theater with a volume of 3,000 m^3 and a seating capacity of 530. It is made up of fixed seating section C delimited by a wall joining mobile seating section D (used only for the "Italian" version), which is inclined by means of a descending floor. Laterally the room is delimited by sliding walls that, at the same time, conceal mobile sections A and B, and it is separated from the stage by a mobile portal that ensures variable opening of the stage horizontally (8-10 m) and vertically (6-8 m) by means of a gangway with lights and the portal (Fig. 1).

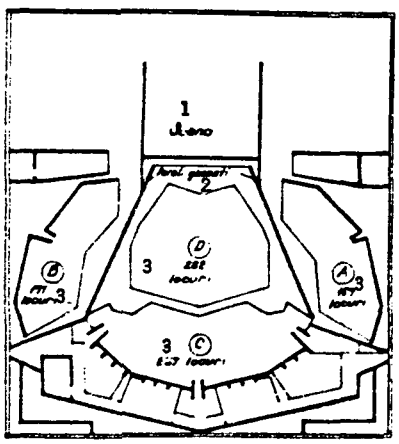

Fig. 1. "Italian" version. 1. Stage; 2. Sliding walls; 3. Seats.

b) An "Elizabethan" theater with a volume of 3,035 m^3 and a seating capacity of 455. This is made up of fixed seating section C delimited by a wall joining mobile seating section B delimited by walls or a panel of interior decoration that constitutes the background of the stage and conceals mobile seating sections A and D. /346

The actors perform on an elongated podium that is placed within the room and consists of passable elements arranged perpendicularly to the scenery, with the audience sitting on the steps of seating sections C and B, which surround the three sides of the podium like a horseshoe (Fig. 4).

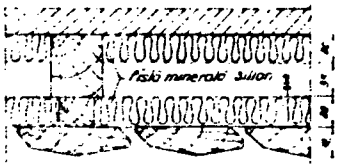

Fig. 2. Acoustic treatment applied to walls of fixed section C (in all versions). 1. Mineral felt.

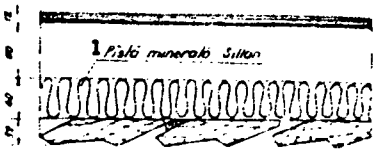

Fig. 3. Acoustic treatment applied to sliding walls for "Italian" version. 1. Mineral felt.

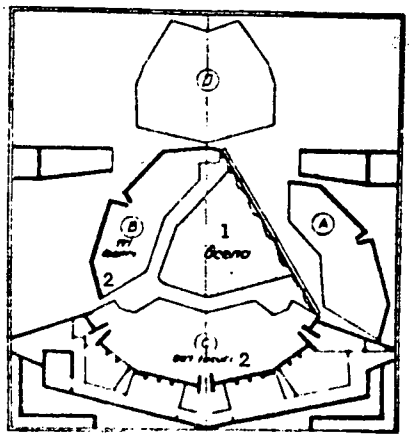

Fig. 4. "Elizabethan" version. 1. Stage; 2. Seats.

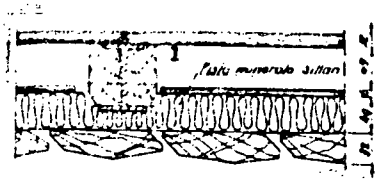

Fig. 5. Acoustic treatment applied to walls of mobile sections A and B for the "arena" and "Elizabethan" versions (one-third of wall surface). 1. Sillan mineral felt.

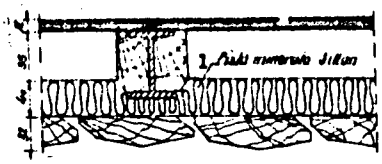

Fig. 6. Acoustic treatment applied to walls of mobile sections A and B for "arena" and "Elizabethan" versions (2/3 of wall surface) 1. Sillan mineral felt.

c) An "arena" theater with a volume of 3,600 m^3 and a seating capacity of 598. This is made up of fixed seating section C delimited by a fixed wall joining the mobile walls of sections A and B delimited by their walls. /347

The actors perform on a central podium consisting of passable elements placed directly on the central descending floor on a level with the floor of the room. The audience sits on the steps of fixed seating section C and of mobile seating sections A and B, which are placed circularly around the arena podium. Mobile section D is placed outside the room on the stage for the "Italian" version (Fig. 7).

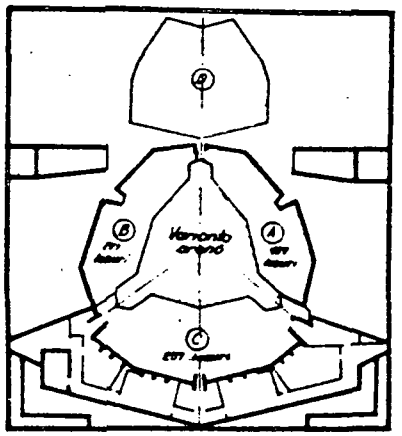

Fig. 7. "Arena" version.

The ceiling of the room consists of a central ceiling which covers the greater part of the room and which, owing to the arena version, is parallel with the basic floor of the room, as well as a number of peripheral ceilings that are continuous with the walls of fixed seating section C and with the walls of mobile seating sections A and B. /348

2. Room Acoustics

The necessity of ensuring the best possible intelligibility of the sounds emitted from the stages of the three room versions toward the audience under the conditions of natural acoustics dictated the resolution of relatively complicated problems because the three room versions have in common only the fixed ceiling of the room, fixed seating section C with the afferent walls and part of the ceiling.

In order to create the acoustic parameters that are necessary for the three room versions we designed, carried out and tested five types of acoustic treatments that were applied to walls and ceiling of fixed seating section C as well as to the sliding walls with their afferent ceilings (Figs. 2, 3, 5, 6 and 8). /349

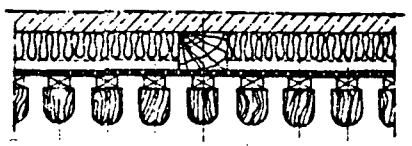

Fig. 8. Acoustic treatment applied to ceilings of fixed and mobile seating sections A, B and D for all versions.

The fixed ceiling of the room is the principal factor in reenforcing the direct sound by the reflected sound, which entails an increase in intelligibility.

In view of the purpose of the room in its three versions, namely, spoken theatrical productions, and knowing that the volume of this room varies with the version from 3,000 to 3,600 m^3, the optimum value of reverberation time at a frequency of 1,000 Hz is 0.95 sec.

As a result of applying the above-mentioned acoustic treatment, for the "Italian" version (the most frequently used) we get a reverberation time of 0.96 sec and for the other versions reverberation time varies from 0.98 sec to 1.06 sec.

Another difficult problem, that of flutter echoes between fixed ceiling and performing podium, cropped up in the case of the two

versions in which the podium is situated within the room, namely, the "Elizabethan" version and, especially, the "arena" version, in which these two surfaces are parallel; this problem was solved by applying differently-sized semispherical bodies to the entire free surface of the ceiling.

3. Noise Reduction

The auditorium is air-conditioned by means of a conditioning plant equipped with fans that generate a high noise level.

By applying noise reducing devices along the airways, we reduced the noise transmitted to the room down to the admissible level (Cz25), which was exceeded by 1-2 dB at low frequencies.

Conclusions

1. The auditorium of the Bucharest National Theater has been /350
equipped with a number of fixed and, especially, mobile structural elements. By maneuvering the mobile elements the room is transformed into three functional types of rooms for three different types of productions: Italian, Elizabethan and arena.

2. In view of the fact that the three versions have only a few elements in common: the fixed ceiling of the room, the fixed seating section with afferent walls and part of the ceiling, the problem of ensuring the optimum reverberation time for the volume of the room necessitated the carrying into effect of five types of acoustic treatment that were applied to the fixed and mobile structural elements.

3. The reverberation time achieved has values that are very close to the optimum for the "Italian" version (0.96 sec), and for the other versions there appear small differences (0.97-1.06 sec), which does not affect the acoustic qualities of the room in their three versions.

4. The problem of flutter echoes between podium and ceiling in the "Elizabethan" and "arena" versions was solved by applying semispherical bodies to free portions of the fixed ceiling.

5. The level of noise transmitted to the auditorium was reduced down to the Cz25 curve.

Summary

The auditorium of the Bucharest National Theater has been /351
equipped with devices permitting it to be transformed into three

functional types of rooms for three different types of production: Italian, Elizabethan and arena.

Because the three versions have very few elements in common, in order to bring about the acoustic parameters that are necessary to ensure the good acoustics of the three room versions, we developed five different types of acoustic treatment.

In view of the purpose of the room (spoken theater) and the volume of the room (3,000-3,600 m^3) the optimum reverberation time is 0.95 sec at a frequency of 1,000 Hz. By applying the different types of acoustic treatment we brought about a reverberation time ranging from 0.96 to 1.06 sec.

The problem of flutter echoes in the arena and Elizabethan versions was solved by applying semispherical bodies to the fixed ceiling.

The level of noise transmitted to the auditorium was reduced down to the Cz25 curve.

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OPTIMIZING ACOUSTICAL TREATMENT

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Sound absorption plays a determinant role in obtaining good /353
acoustics for theater auditoriums. Next to laying out profiles, the choice of sound-absorbing materials and structures which, in addition to stability, resistance in time and a handsome appearance, possess a number of sound absorption coefficients of a given value, constitutes the basis of acoustical design.

According to L. L. Beranec [1], sound absorption determines reverberation time and its variation on frequency bands, influences echoes, the distribution of the sound energy and the incidence of the first reverberated wave, the intelligibility, the clarity and the sonority of the room.

Design starts from the volume of the room, the surface of the walls, ceiling and free floor and the number of spectators. From tables or nomograms (Fig. 1) the suitable reverberation time for the volume and purpose of the room is chosen. Knowing the law of variation of the reverberation time, curves of permissible reverberation times are plotted as a function of frequency for the room in question (the example of Fig. 2).

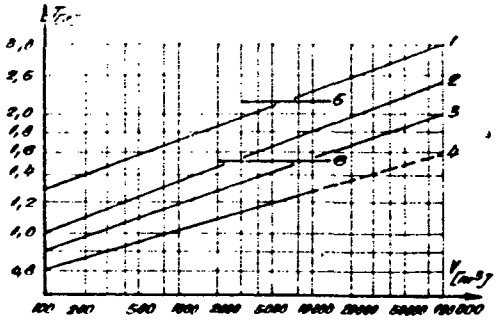

Fig. 1.

Sabine's relation T = kV/A permits direct determination of the total absorption necessary for the room.

It is known that the necessary absorption can be brought about by a great number of variants, using different combinations of materials and structures each of which have a different series of sound absorption coefficients.

It is obvious that not every version is acceptable from the /355
standpoint of aesthetics, resistance and economic efficiency. Inasmuch as calculation is laborious, in practice it is the custom to work out two or three versions and, by successively modifying the proportions of the elements, to reach a reverberation curve that falls within the limits indicated in Fig. 2.

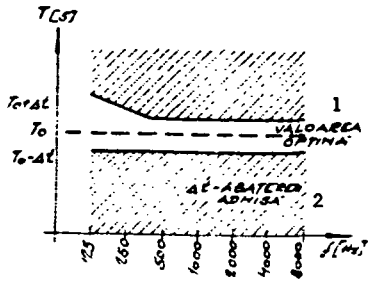

Fig. 2. 1. Optimum value; 2. Admissible deviation.

The method proposed in the present study uses a linear programming model that permits the architect to choose with full knowledge of the technological and economical indexes (Fig. 3) a solution from among a great number of automatically calculated versions.

1. Problem

We propose to determine the surfaces X_j, $j = 1, 2, \ldots, n$ occupied by the material elements M_j that determine the acoustic parameters of an auditorium with given geometrical characteristics and which, by satisfying a set of conditions K_i, $i = 1, 2, \ldots, n$, minimize the cost of the treatment.

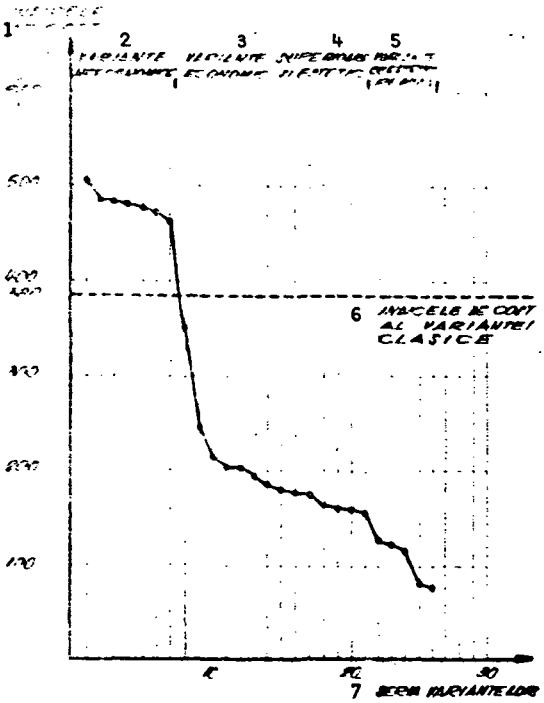

Fig. 3. 1. Cost indexes; 2. Uneconomical versions; 3. Economical versions; 4. Aesthetically superior versions; 5. Less aesthetic versions; 6. Cost indexes of classical version; 7. Series of versions.

2. Initial Data

Let $M = \{M_k\}$, $k = 1, 2, \ldots, N$ be the set of material elements that determine the acoustic parameters of the room. With each element M_k is associated the vector $(a_{1k}, a_{2k}, \ldots, a_{mk})$ whose components represent the coefficients of the relations for the conditions that come into play in formulating the linear programming model.

From set M we select the subset of acoustically compatible elements $N = \{M_j\}$ that can be employed in the type of treatment under consideration. This will result in different calculation hypotheses for "hot" treatment of woodwork and draperies or "cold" treatment, as in the case of simple joining of concrete, bricks, plasters, etc. In each of these cases by means of appropriate reordering we shall regard the set N as being $N = \{M_j\}$, $j = 1, 2, \ldots, n$ where obviously $n < N$.

We regard as data the total surface of the room S_T, the partial surfaces of the ceiling, stage, free floor, floor occupied by seats, space under the balcony, etc., as well as the total optimum absorption of the room for each value of the chosen frequency range. /356

3. The Restrictions of the Model can be grouped into acoustic, geometrical, aesthetic restrictions regarding the extent of occupation of the room and, obviously, non-negativity.

3.1. Acoustic restrictions. To each of the q values chosen in a frequency range there corresponds for each element M_j an acoustic absorption coefficient a_{ij}, $i = 1, 2, \ldots, q$, $j = 1, 2, \ldots, n$. Denoting by δ_i the admissible deviation of the total absorption A_i for the respective frequency, the acoustic conditions can be formulated mathematically as follows:

$$\left| \sum_{j=1}^{n} a_{ij} x_j - A_i \right| \leq \delta_i, \quad i = 1, 2, \ldots, q \leq m \tag{1}$$

3.2. The geometrical restrictions imposed on the variables refer to the manner of covering given surfaces.

It is obvious that we must have

$$\sum_{j=1}^{n} x_j \leq S_T \tag{2}$$

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Likewise imposed are conditions of the form

$$x_{j'} \leq S_p, \quad j' \in (1, 2, \ldots, n), \quad p \in (q+1, \ldots, m) \tag{3}$$

in order to concretize the partial surfaces leading to the stage, orchestra pit, etc.

Also necessary are conditions that determine the covering of parts of the room with certain compatible materials such as, for example, the covering of doors, of the free floor, of the ceiling, etc., and that can be expressed in the form of equations such as

$$\sum_{j=1}^{n} x_j \leq S_p, \quad p \in (q+1, \ldots, m) \tag{4}$$

3.3. Aesthetic restrictions. The aesthetic restrictions that can be introduced into a linear programming model refer to the selection of asethetically compatible materials that can or must cover certain surfaces. The choice of coefficients can be established by the architect himself who wishes to influence the composition of the acoustic treatment.

Such restrictions can be formulated either as the upper or lower limits of certain variables

$$x_j \leq b' \quad \text{or} \quad x_j \geq b'' \tag{5}$$

or in the form of proportional determinations

$$a_{ij_1} x_{j_1} - a_{ij_2} x_{j_2} \cdot o \qquad (6)$$

or, under extreme conditions, as an equality by means of which the surface covered with the material chosen by the architect is predetermined. /357

3.4. Restrictions concerning the extent of occupation by spectators is introduced in the form of the following relation

$$a_{ij'} x_{j'} + a_{ij''} x_{j''} = S_p; \quad a_{ij'} + a_{ij''} = 1 \qquad (7)$$

where j' and j" correspond to occupied and empty seats, respectively, and the coefficients a_{ij} indicate the corresponding extent of occupation.

3.5. Restrictions of non-negativity. Since we are talking about variables that signify covered surfaces they must not be negative, i.e.,

$$x_j \geq 0 \qquad j = 1,2, \ldots n \qquad (8)$$

4. Objective Function

The model permits adoption of different optimization criteria. Total cost of the acoustic treatment (cost of manual labor and execution time) can be minimized. In all these cases, by associating the proper index c_j to each variable x_j we get the objective function

$$f(x) = \sum_{j=1}^{n} c_j x_j \longrightarrow \text{minim} \qquad (9)$$

5. The classical model of linear programming is obtained by combining relations of the type of Eqs. (1)-(9), which can be solved by known methods.

6. Results and Conclusions

The mathematical model we have briefly described above was tested by the Cluj Institute of Design for Town Planning and Building on the occasion of designing the Lugej Union Clubhouse and the Timisoara Youth House, calculations being carried out on the IPROTIM's MECIPT-2 electronic computer.

Having considered about 50 materials for each type of treatment we obtained over 30 solutions which, tending toward the

optimum solution, improve successively economical indices and acoustic treatment. Applying the model led to a reduction in the cost price on the order of 150,000 lei. /358

Also to be noted in addition to the obvious economical efficiency that can be achieved by applying such a model are the reduction in designing time, the saving of human effort and, especially, the possibility that the designer has of choosing, depending on his aesthetic concept, a structure out of those in the optimum region, with complete knowledge of the deviation from minimum cost.

More than this, through post-optimization the methodology makes possible a restriction-changing dialogue between designer and calculator. This permits analysis and solution of acoustic treatment with the designer's creative participation, with a view to new solutions.

At the present time, extension of this model is being studied by introducing more pretentious aesthetic conditions which, however, from the mathematical point of view implies approaching the problem as a nonlinear programming model.

Summary

A mathematical linear programming model is presented for /359 optimizing acoustical treatment and interior decoration of concert and other public halls. This method provides the designer with a range of acoustically correct solutions at increased economical efficiency.

The mathematical model uses geometrical data about the room, recommended reverberation time values, the architect's choice of given sound-absorbing structures and finishing materials. The model permits inclusion of aesthetical considerations about conditioning, proportioning, or, on the contrary, reciprocal exclusion of any classes of material and/or sound-absorbing structure.

Owing to the range of solutions experimental application of the model proved to be of remarkable economical efficiency from the standpoint of costs, shortening of execution time and saving of design effort.

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NUMERICAL COMPUTATION OF ECHOGRAMS IN A ROOM: APPLICATION TO DETERMINATION OF INTELLIGIBILITY

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Summary

For many years acousticians have used echogram recordings to estimate the acoustic quality of a room. Using two methods based on geometric acoustics, we have computed echograms for rooms of irregular shape. The image method and the ray method have been applied to a heptahedral room as well as a conference hall. The results obtained, which have been verified experimentally, permit an estimate of the intelligibility coefficient.

/361

1. Introduction

In acoustics considerable significance is generally attached to echograms obtained by emission of an extremely brief acoustic pulse in a room. Recordings taken at various sites within the room permit a fairly accurate indication of the order of arrival of the different echoes attendant on the direct sound, with their respective intensities. It seemed advantageous to us to be able to obtain such echograms by numerical computation, since this would permit advance knowledge of the acoustic quality of a room to be constructed without the necessity of building a model. Experimental verification has made it possible to determine the degree of reliability of these computations based solely on geometric acoustics.

2. Geometric Acoustics Methods

Let us consider a source S of energy W, located within the room. Within a solid angle component $d\omega$ the energy emitted is:

$$dW = I(\theta, \varphi)\, d\omega$$

if $I(\theta,\phi)$ is the intensity of emission in the direction indicated the spherical coordinates θ and ϕ. The corresponding energy reaching the receptor is;

$$dw = s(\theta, \varphi)\, dW .$$

$a(\theta,\phi)$ is an attenuation factor equal to unity for a ray proceeding directly from the source to the receptor, if absorption by the air is not taken into account; more generally, this term is equal to a product of factors; it may be nil if the ray is completely absorbed by a given wall of the room before arriving at the receptor.

One of the methods permits computation of the illumination at point C, the center of the receptor; and the other, the average illumination. /362

2.1. Image Method

Let us consider a polyhedral room: the source possesses images in relation to the various walls, and each of these images possesses images in turn: from reflection to reflection, it is possible to arrive at a large number of image sources. Rather than considering the images of the source, one may consider those of the receptor, each of which is perceived from the source at a given solid angle $\Delta\omega j$.

In the case of a spherical receptor:

$$\Delta\omega j = A/4Rj^2$$

if $A = 4\pi r^2$ is the area of the receptor of ray r, and Rj the distance from the source image to the center C of the receptor. If the receptor is flat:

$$\Delta\omega j = Ahj/Rj^3$$

if hj is the distance from Sj to the plane of the receptor, whose dimensions are assumed to be smaller than those of Rj. If:

$$\Delta Wj = I(\theta j\ ,\ \varphi j)\ \Delta\omega j$$

is the energy transmitted by slab $\Delta\omega i$, the corresponding energy received will be:

$$\Delta wj = I(\theta j,\ \varphi j)\ a(\theta j,\ \varphi j)\ \Delta\omega j$$

and the total energy received:

$$w = \sum_j A.I(\theta j,\ \varphi j)\ a(\theta j,\varphi j)/4Rj^2$$ for a spherical receptor

and

$$w = \sum_j hj\ I(\theta j,\varphi j)\ a(\theta j,\varphi j)/Rj^3$$ for a flat receptor.

There will be no discussion here of the ordinary case of a

rectangular room, for which it is very easy to determine the coordinates of the image sources. For a given polyhedral room, the coordinates of the sources must be computed according to their order, that is, their number of reflections, and those which are perceived by the receptor are retained [1].

2.2. Ray Method

The source S is assumed to emit rays in directions selected at random. Each ray, carrying a given quantity of energy depending on the characteristics of the source, may reach the receptor directly or after successive reflections, or on the other hand it may first be absorbed. If N rays have been defined, the energy received will be:

$$w' = \sum_{i=1}^{N} I(\theta i, \varphi i)\, a(\theta i, \varphi i)\, \Delta\omega' i$$

$\Delta\omega' i$ being the solid angle associated with a ray of direction θi, ϕi.

2.3. Comparison of the Two Methods

In the image method the receptor is assumed to be concentrated in O, whose illumination w/A is computed by this method. The ray method makes it possible to determine the amount of energy received by area A. Theoretically the results will not be identical. /363

To represent the energy received as a function of time, the distances from the image sources to the receptor, or the length of the rays received, are classified in increasing order of size. If t is said to = R/c, where c is the wave velocity of the sound, echograms will be obtained in the coordinate plane (t, Δw). These may be used to determine the cumulative energy curves:

$$w(t) = \sum_{j' \le j} \Delta w_{j'} \quad \text{for} \quad t_j \le t < t_{j+1} \qquad \text{(image method)}$$

and

$$w'(t) = \sum_{i' \le i} \Delta w'_{i'} \quad \text{for} \quad t'_i \le t < t'_{i+1} \qquad \text{(ray method)}$$

In order to compare curve $\Delta w_j(t_j)$ with curve $\Delta w'_i(t'_i)$, let us consider the images Mj of the receptor seen from S at solid angle $\Delta\omega_j$. The solid angles $\Delta\omega_j$ of the closest images, on the average, are the largest, and may contain several slabs $\Delta\omega' i$ used in the ray method; for low values for t, the $\Delta\omega' i$ factors are smaller but more concentrated than the Δw_j factors; the opposite is true for large values for t.

3. Results for a Heptahedral Room

The first example is an irregular seven-sided polyhedron; the walls are vertical and the ceiling oblique; the volume of the room is 173 m^3. It has been assumed that the floor coating is a completely absorptive material, that the reflection coefficient of the walls is independent of the incidence of the rays and equal to 0.9, and that the isotropic source and the receptor have a surface area of 1 m^2.

Fig. 1 shows the curves obtained by the image method, limiting the computation to the 4°, 5°, 6° and 7° orders. It appears that a curve limited to a given order is accurate to the extent that it merges with that limited to the next higher order. Furthermore, the similarity of the curves increases as the maximum order is increased. The time required for computation made it necessary to limit the procedure to a relatively low order; in fact it may be estimated that the number of image sources whose coordinates must be computed is an exponential function of the order. Although the image source method is well suited for computing close echoes, it is very poorly adapted to the problem of distance reverberation.

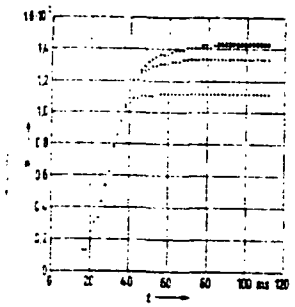

Fig. 1.

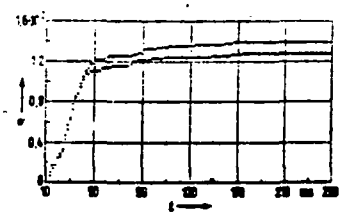

Fig. 2.

Fig. 2 shows the results obtained by the ray method, using the same theoretical positions for the source and the receptor. Two different printings of 4000 emitted rays were used; the difference between the two curves indicates the degree of error in the method. It may be noted that this method of computation shows no error when the values for t are high. The curves in Figs. 1 and 2 are very similar [1]. /364

4. Results Obtained in a Conference Hall

The conference hall for the Laboratory Group of Marseille of the CNRS (Centre national de la recherche scientifique; National Scientific Research Center] is a room with a capacity of

176 which we considered to be delimited by five parts of cylinders with vertical or horizontal axes, and three flat walls. The reflection coefficient of the walls was assumed to be independent of the incidence of the rays, and this coefficient was determined experimentally by means of wave trains of 1000 Hz, the length of the experimental pulse being approximately 1 ms.

Since the walls of the room were curved, the ray method seemed to be best suited for use in obtaining echograms by computation.

The computations made, assuming the source to be isotropic and the receptor to be spherical, were verified experimentally. The response p(t) of the room to pulse excitation was recorded, and a curve w(t) was obtained by evaluating the integral:

$$w(t) = \int_0^t p^2(t)\,dt$$

Fig. 3 shows the results of computation and those of the experiment for two receptor sites in the room under consideration. These results may be considered to be in satisfactory agreement, given both the imperfection of the experimental pulse and the simplification of the shape of the room made to avoid overburdening the computations.

It should be noted that echograms plotted in this way represent only part of the results: the direction of the echoes is not taken into account here, and this is not a negligible problem.

5. Estimate of Intelligibility

The previously computed cumulative energy curves may be used to obtain an estimate of intelligibility. Let us assume that the useful energy of the direct sound is:

$$E_u = \int_0^{50\text{ms}} p^2(t)dt$$

and the detrimental energy, with a disturbing effect: /366

$$E_N = \int_{50\text{ms}}^{\infty} p^2(t)dt \qquad [2]$$

A signal/noise ratio may be defined which is equal to E_u/E_N. If the level of direct sound is known, it is possible to evaluate the E_u and E_N levels in decibels. A system of experimental curves may then be used to furnish the intelligibility coefficient as a function of the useful sound level, with the detrimental sound level as a parameter [3]. From this one may obtain the intelligibility coefficient of the room at the site considered.

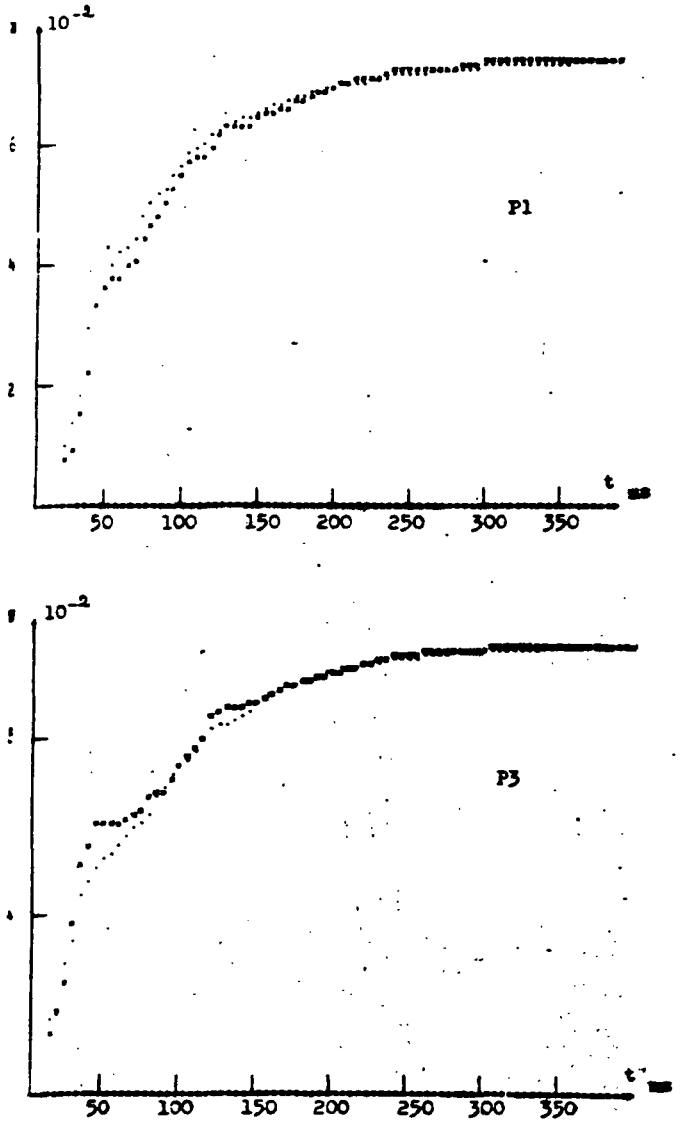

Fig. 3.

For the conference hall the direct sound level was assumed to be 60 dB at 1 m. The coefficients determined for sites P1 and P3 were 90% and 93% respectively.

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N74-28120

NEW THEORY ON THE REVERBERATION OF ROOMS

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The reverberation time of a room basically depends on two factors: the mean free path of sound between two successive reflections on the walls and the absorption which the sound wave undergoes at each reflection. The theories for calculating reverberation time which have been developed so far assume: /369

-- either that the sonic energy loss takes place in continuous fashion, causing the sudden energy loss occurring at each reflection to be somewhat distributed over the entire mean time T taken by the sound wave to travel from one wall to another; this is the Sabine formula:

$$T_r = 0{,}16V / \Sigma \alpha_i S_i \qquad (1)$$

-- or that this energy loss is discontinuous from second to second but is always the same at each reflection: at a given instant, the energy becomes $E_{nt} = E_{(n-vt)}(1-\bar{\alpha})$, $\bar{\alpha}$ being the mean absorption coefficient, that is,

$$\bar{\alpha} = (\alpha_1 S_1 + \alpha_2 S_2 + \cdots + \alpha_n S_n)/S \qquad (2)$$

The Eyring formula is thus obtained:

$$T_r = 0{,}07V / S \log (1-\bar{\alpha})^{-1} \qquad (3)$$

-- or that this discontinuous energy loss is indeed the same at each reflection, but is such that:

$$E_{nr} = E_{(n-1)r}(1-\alpha_1)^{x_1}(1-\alpha_2)^{x_2}\cdots(1-\alpha_n)^{x_n}$$

$x_1, x_2 \ldots x_n$ being the ratio of the surface areas $S_1, S_2 \ldots S_n$ of absorptive material to the total surface area of the walls:

$$x_1 = \frac{S_1}{S}, \quad x_2 = \frac{S_2}{S}, \quad \ldots\ldots\ldots x_n = \frac{S_n}{S}$$

And the Millington formula is obtained:

$$T_r = 0{,}07\,V \Big/ \Sigma S_i \log(1-\alpha_i)^{-1} \quad (4) \qquad (4)$$

All the calculations made to obtain these formulas, however, /370 are based on a value for the mean free path equal to 4 V/S, determined by using regularly distributed sound rays from a point source and studying the path of the rays across the room without attempting to make them pass by the listening point where the reverberation should be perceived. These calculations also assume that sound travels in flat waves, since the energy loss takes place only at the instants of reflection from the walls and not, for example, by dispersion by spherical waves.

On the basis of the idea that this mode of propagation should examined and that a study should be made of all the paths of rays traveling from a source point to a listening point, we have constructed a new theory termed the theory of image sources.

I. Theory of Image Sources

This theory was established for a parallelepipedal room. At a listening point the sound waves seem to originate from a number of sources which include, in addition to the actual source, the image sources of the latter in relation to the six surfaces of the room. We have shown [1] that these images, corresponding to reflections of the nth order, are located in the vicinity of the

surfaces of an octahedron whose apices can easily be positioned in relation to the center of the room. The number of these apices is equal to $4\,n^2 + 2$. The situation at the listening point is just as if all the sound rays were originating simultaneously from all the image sources having a virtual acoustic power which decreases in proportion to the increases in the nth order of the reflections.

The image sources are regularly distributed throughout the room, so that sound rays traveling from the source to the listening point after numerous reflections on the walls undergo the same number of impacts on each of the latter, that is:

$$\frac{n(n+1)(n^2+n+1)}{6} \qquad (5)$$

These are fairly regularly distributed on the room surfaces if the source point and the reception point are not close to the walls. This figure increases fairly rapidly as a function of n.

Tracing of these sound rays and evaluation of the density of the reverberated energy and the reverberation time at the listening point led us to define two new types of mean free paths [2].

The first type: l_m

$$l^2_m = \frac{4n^2+2}{n} \cdot \frac{1}{\sum A_n l_{in}} \quad (n \to \infty) \qquad (6)$$

l_{in} being the distance from an image source of the nth order to a listening point. This factor is used to calculate the intensity of reverberated sound at the listening point.

The second type: l'_m

$$l'_m = \frac{1}{n}\sum_{i=1}^{4\pi r^2} l_{in}/(4\pi r^2) \qquad (7)$$

is the mean free path used to calculate the reverberation time at the listening point.

II. Calculation of the Intensity of Reverberated Sound at the Listening Point /371

To make a precise calculation it is necessary to know the position of the image sources, the distance which separates each of them from the listening point, and the acoustic power which seems to originate from each image source. This acoustic power can be exactly determined only if the positions of the points of impact on the walls are marked and the absorption coefficients of the wall coatings at these points are known. This calculation can be performed on a computer without difficulty, but two approximations made by means of a simpler computation technique should give an approximate result.

1. Since the number of impacts is relatively high, beginning with the 3rd order of reflection, and since these points are fairly regularly distributed over each surface, it is possible to estimate that the number of these points found on a given material in relation to the total number of impacts for a given path of the ray from the source point to the reception point is equal on the average to the ratio S_1/S of the surface area of the material to the total surface area of the walls. This approximation will be all the more valid if the material is well distributed in small components over all the surfaces.

The power from a source of the nth order may be considered equal to:

$$W(1-\alpha_1)^{n x_1}(1-\alpha_2)^{n x_2} \cdots (1-\alpha_n)^{n x_n} \tag{8}$$

or $$W\left[(1-\alpha_1)^{x_1}(1-\alpha_2)^{x_2} \cdots (1-\alpha_n)^{x_n}\right]^n$$

2. The second approximation consists in assuming that all the image sources of the nth order are the same distance from the listening point, that is $n l_m$.

The contribution of all these sources of the nth order to the intensity of the reverberated sound is thus:

$$I_{Rn} = (4n^2+2) W\left[(1-\alpha_1)^{x_1} \cdots (1-\alpha_n)^{x_n}\right]^n / 4\pi n^2 l_m^2 \tag{9}$$

and the total intensity of the reverberated sound will be:

$$I_{RT} = \sum_{n=1}^{\infty} I_{Rn} \tag{10}$$

An adequate approximation of the value for I_{RT} will be:

$$I_{RT} = Wq / n l_m^2 (1-q) \tag{11}$$

assuming that

$$q = (1-\alpha_1)^{x_1} \cdots (1-\alpha_n)^{x_n}$$

When the absorption coefficients α_{19}, α_{21}, ... α_n are sufficiently close to the absorption coefficient $\bar{\alpha}$ such that

$$\bar{\alpha} S = \alpha_1 S_1 + \cdots + \alpha_n S_n \tag{12}$$

it may be stated that $q \approx (1-\bar{\alpha})$ and an expression is found for the intensity of the reverberated sound I_{RT} which we have already given in a previous study [1].

$$I_{RT} = W(1-\bar{\alpha}) / n l_m^2 \bar{\alpha} \tag{13}$$

III. Calculation of Reverberation Time at the Listening Point /372

In examining how the total intensity of reverberated sound at the listening point decays in stages once the source has been stopped, we have shown [3] the configuration of the sound decay curve: this is in the form of a descending stairway made up of a large number of small, irregular steps.

The remaining intensity level at the listening point after the drop in energy corresponding to reflections of the nth order will be

$$\frac{W}{\pi l_m^2}\frac{q}{1-q} - \frac{Wq(1-q^n)}{\pi l_m^2(1-q)}$$

that is,

$$\frac{W}{\pi l_m^2}\frac{q^{n+1}}{1-q} \qquad (14)$$

The drop in level of intensity of the reverberated sound thus falls on the average between the two suspensions of sonic energy production due to two orders of neighboring reflections:

$$10\log\left(\frac{Wq^{n+1}}{\pi l_m^2(1-q)}\Big/\frac{Wq^{n+2}}{\pi l_m^2(1-q)}\right) = 10\log q^{-1} \qquad (15)$$

The length of time of reverberation will correspond to a drop of 60 dB.

$$10n\log q^{-1} = 60 \quad , \quad n = 6/\log q^{-1} \, , \quad T_r = n\tau$$

T = the length of time required for the sound to travel a path of length l'_m.

$T_r = 6l'_m/c\log q^{-1}$ which can be put in another form:

$$\boxed{T_r = \frac{6l'_m S}{c\sum S_i \log(1-d_i)^{-1}}} \qquad (16)$$

The expression found is fairly close to that proposed by Millington. The formulae would be exactly the same if one assumes the value 4 V/S for the mean free path l'_m, that is:

$$T_r = \frac{0,07\,V}{\Sigma S_i \log(1-\alpha_i)^{-1}} \qquad (17)$$

If the α_i factors do not differ excessively from the mean absorption coefficient, the reverberation time may be considered equal to:

$$T_r = 6 l'_m / c \log(1-\alpha)^{-1} \qquad (18)$$

a formula which is close to that of Eyring, if l'_m is not too far from 4 V/S.

The limit of the mean free path l'_m for $n \to \infty$ has been calculated by Nguyen Tho Tuong [4]. We have given a closely similar expression. /373

$$l'_m = \frac{1}{6}\left[(a^2+b^2)^{1/2}+(b^2+c^2)^{1/2}+(c^2+a^2)^{1/2}\right]$$

where a, b, and c are the three dimensions of the room.

Two applications of this new theory have consisted in measuring the absorption coefficient in a reverberating room and the response of a room to sonic pulses.

Summary /374

The inadequacy of the various theories which have been proposed for finding the reverberation time of rooms can be explained by an attempt to examine what might occur at a listening point when image sources of determined acoustic power are added to the actual source. The number and locations of the image sources are stipulated. The intensity of sound at the listening point can be calculated by means of approximations whose conditions for

validity are given. This leads to the proposal of a new expression for the reverberation time, yielding results which fall between those obtained through use of the Eyring and Millington formulae; these results are made to depend on the shape of the room by means of a new definition of the mean free path.

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N74-28121

ANECHOIC CHAMBER IN INDUSTRIAL PLANTS

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Acoustical measurement and analysis of noise require an anechoic chamber that will reproduce the conditions of a free acoustic field. /375

The very strict conditions imposed on anechoic chambers intended for basic research justify the relatively high cost of attaining a reduced environmental noise level and releasing stationary waves down to low frequencies.

In the case of an anechoic chamber in an industrial plant it seems reasonable to keep the imposed conditions to a bare minimum, especially in the matter of soundproofing, because the noise sources that are studied generate and radiate high or very high acoustic pressure levels.

What is called for is that for each separate case the minimum conditions be specified that an anechoic chamber in an industrial plant must meet so as to optimize its construction and ease of handling, the necessary space and execution costs.

These conditions can be summarized as follows:

a) background (environmental) noise must be at least 10 dB below the measurement range;

b) the anechoic chamber must be big enough to permit measurements on the largest machines used in the plant's production;

c) walls and ceiling of the anechoic chamber must consist of modules assembled on the spot;

d) the anechoic chamber must be assembled with current tools and without preparing a special foundation. /376

Under these conditions, in principle, no difficulties appear in using a light-weight anechoic chamber.

Building an Anechoic Chamber in an Industrial Plant

The anechoic chamber was designed and built for a plant that manufactures implements and equipment for the textile industry.

As in the case of the classical design, the anechoic chamber consists of two housings: an outer sound-insulating housing and an inner sound-absorbing housing.

The space between the two housings is nonuniform in the vertical as well as the horizontal direction, which contributes to efficient absorption in a wide band.

Requirements with respect to the transmission loss TL are modest. It can be seen from the diagram of Fig. 1 that the environmental noise level in the assembling room in which the anechoic chamber is to be mounted is rather low; in this case the outer housing, which consists of plastic modules, provides enough damping, TL being greater than 10 dB. The intensity level necessary for noise measurements turns out to be far enough removed from the background noise level. The vibrations transmitted through the floor are likewise insignificant in amplitude and the chamber can, therefore, be mounted directly on the floor.

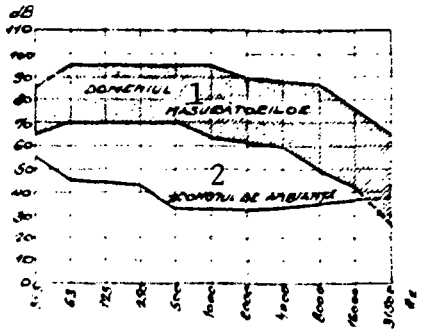

Fig. 1. 1. Measurement range; 2. Environmental noise.

The outer housing consists of self-supporting wall and roof modules made of glass fiber reinforced polyester resins. A 20 mm thick layer of mineral felt is applied to the inner surface of the modules during the poly spray process. The wall modules are shown in Fig. 2 (seen from behind, the side and in sections). Module thickness is 3 mm; owing to its sectional shape and stiffening plates we get sufficient stability. /377

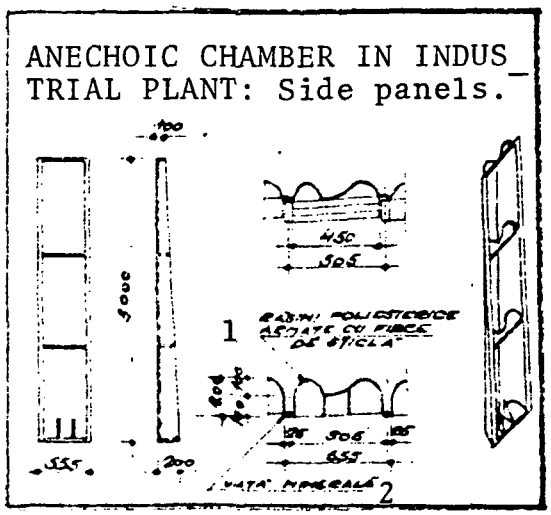

Fig. 2. 1. Glass fiber reinforced polyester resins; 2. Mineral cotton.

Figure 3 shows the shape of the modules and the manner of building the inner housing. Each module has a total of 3 x 6 = 18 pyramidal wedges, and by placing the modules in one direction and in another we obtain the diffusion necessary for the acoustic field.

Figure 4 shows how the anechoic chamber is assembled; note the wall modules and the ceiling, which consists of type III (roofing) panels also made of glass fiber reinforced polyester resins. The chamber is assembled by means of mechanical screws or magnetic coupling.

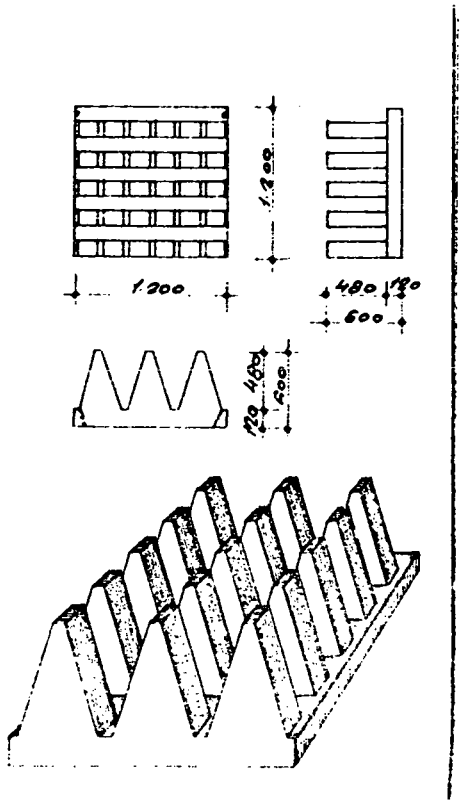

Fig. 3. Anechoic chamber in industrial plant: absorbing modules (wedges).

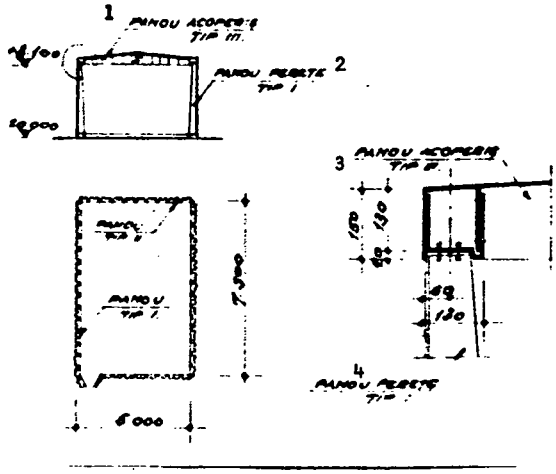

Fig. 4. Anechoic chamber in industrial plant: assembly. 1. Type III roofing panel; 2. Type I wall panel; 3. Type II panel; 4. Type I panel.

The inner dimensions are 7.500 x 5.000 m and the useful height is close to 3 m.

By virture of its size this anechoic chamber belongs to the class of small measurement chambers but it is large enough for measuring and analyzing the noise of textile machines of current manufacture, either as a whole or by subassemblies.

Summary

The paper reports on a light anechoic chamber for routine acoustical measurements in the machine-building industry. /379

The outer housing of the chamber consists of modules cast in glass fiber reinforced polyester resin; the inner housing consists of pyramidal modules cut out of sound-absorbing slates.

The parameters of this anechoic chamber facilitate acoustical measurements according to ISO and CAEM recommendations.

Advantages: reduced weight, encumberance and cost; possibility of assembling in short time; when not in use, the modules can be stored, saving productive floor area.

N 74-28122

ASPECTS OF CHECKING INSTALLATIONS FOR THE NOISE LEVEL OF ELECTRONIC EQUIPMENT

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One of the parameters that define the quality of electrical /381
engineering equipment and its manufacturing technology is the level of the noises generated in running.

The problem of noise level reduction has recently acquired particular importance in the context of measures to reduce pollution.

Of especial interest are electrical machines and transformers, for which the latest standards stipulate obligatory tests and noise level measurements, and that these be conducted separately for each product.

Electrical machine noises are caused by the running of moving parts (bearings, grippers, shaft vibration, etc.), as well as by vibrations of the machine's magnetic circuits.

In transformers, noises are caused especially by the upper harmonics of the electrodynamic forces in the sheets as well as in the winding. Moreover, the transformer tank makes its own contribution to the noise level as well to the noise frequency spectrum.

Another category of electrical engineering products, such as horns, bells and other sound alarm devices, require that the fre- /382
quency spectrum as well as the level of the sound generated be ensured.

Radio and TV receivers also must be checked in soundproofed enclosures in order to be able to determine the volume and spectrum of the sounds emitted.

A special role in diversifying and improving electrical engineering products is played by the methods and devices used for noise level measurement in order to furnish the necessary elements for introducing corrections into designs and manufacturing technologies.

The enclosures in which noise is measured must, in principle, meet the following conditions:

-- damp outer noises down to a level where they will not interfere with measurements;

-- eliminate reflections from the interior in order to make it possible to measure progressive acoustic waves.

The basic idea behind fitting out such an enclosure is for the measurements to be conducted in a free field.

In practice, however, the conditions of a free field cannot always be realized, and measurements in reverberating and semi-reverberating acoustic fields are also accepted.

In fitting out the enclosures for measurement of electric machine noise two extremes are to be encountered: in some cases the enclosure is equipped for complete soundproofing and total elimination of reflections while in other cases a completely unequipped enclosure is employed.

The conditions imposed on enclosures in which electric machine and transformer noise is measured, as well as the manner in which the room's performance is determined, depend directly on the desired accuracy and the nature of the measurements. /383

Measurements for the defectoscopy of electric machines by noise analysis must be conducted in a free field, i.e., the following relation must be satisfied:

$$I_2 = I_1 \left(\frac{r_1}{r_2}\right)^2 \qquad (1)$$

where I_1 is the acoustic intensity at distance r_1 from the source and I_2 is the acoustic intensity at the distance r_2 from the source.

Between acoustic intensity levels L_1 and L_2 there must exist the relation

$$L_2 = L_1 + 20 \log \frac{r_1}{r_2} \qquad (2)$$

The ideal free field consists of a perfectly quiet enclosure in which the equipment whose noise is being measured is suspended at a considerable distance from the ground.

Inasmuch as the ideal field cannot be realized in practice, enclosures are employed in which the conditions are as close as possible to the ideal.

In order to conduct measurements it is necessary to know the rules for totaling acoustic levels by totaling background noise and reflected waves with the noise generated by the equipment at the measurement site.

Thus, during the running of a piece of equipment or a machine, /384 the resultant noise level is given by the relation

$$(M_3) \quad L = L_1 + \Delta L$$

where L_1 is the machine noise plus the background noise of the enclosure; ΔL is determined from the curve of Fig. 1 as a function of $L_1 - L_0$ where L_0 is the background noise of the enclosure.

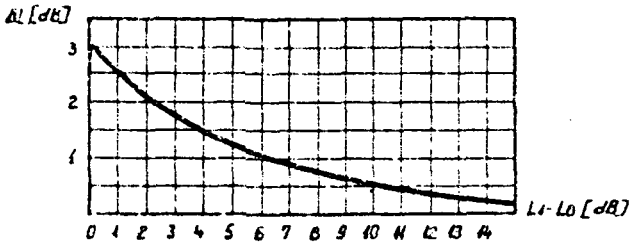

Fig. 1. $\Delta L = f(L_1 - L_0)$ when $L_1 \geqslant L_0$.

It can be seen from Fig. 1 that for $L_1 - L_0 \geqslant 10$ dB, $\Delta L < 0.5$, remaining practically constant. It follows that the level of background noise throughout the spectrum must be kept lower by 10 dB than the level of the noise generated by the running machine.

In the case of less accurate measurements, a difference between L_1 and L_0 smaller than 10 dB is also assumed, but the minimum value is limited to 4 dB on condition that some corrections /385 are made by judiciously placing the microphone near the machine for the purpose of reducing the enclosure's background noise $L_1 - L_0$. It follows from Eqs. (M_2) and (M_3) that an acoustically reflecting obstacle placed in a free field near the measurement site does not interfere with measurements so long as the path traversed by the incident and reflected wave is greater than double the distance between machine and microphone. It follows from Eq. (M_2) that doubling the distance between machine and microphone leads to a reduction in the noise level at the measurement site by 1 dB.

Under the above conditions the reflected wave does not affect the total noise level; measurements can, therefore, be conducted with sufficient accuracy.

The distance between microphone and noise source depends on:

-- the size of the anechoic chamber;

-- the level of the background noise;

-- the size and shape of the noise source;

-- the acoustic field generated by the source.

Taking into account the fact that the distance between the microphone and the walls of the enclosure must be greater than a

quarter of the longest acoustic wave length of interest, the distance between source and microphone is limited in what follows to the size of the anechoic chamber.

On the hypothesis of a high level of background noise the microphone-source distance can be so reduced that the predominant noise at the measurement site will be that generated by the machine. There is, however, the danger that in the event of placing the microphone too close to the machine local values of a nonuniform acoustic field will be measured, which will lead to a deformed picture of the acoustic field.

As a result of the foregoing, in the case of small and /386
medium-sized machines and transformers the distance between microphone and noise source must be on the same order of magnitude as the greatest size dimension of the machine or transformer, but at least 0.3 m.

In the case of larger machines and apparatus the distance between microphone and source must be greater than the longest wavelength corresponding to the lowest frequencies of interest.

We have classified the above desiderata in terms of equipment size.

Studies we conducted led to four different solutions for noise measurement, namely:

1. 5 x 5 m anechoic chambers with free inner dimensions of about 3.2 m in which to measure the noise generated by small sources, such as automobile equipment, acoustic signalling apparatus, etc. (Fig. 2).

2. 6 x 6 m anechoic chambers with free dimensions of about 4.5 x 4.5 m, intended for measuring the noise of small and medium-sized electric motors (1-2 MW) (Figs. 3 and 4).

Anechoic chambers intended for measuring the noise generated by higher powered motors and electric transformers of up to 1.6 MVA, inclusively.

3. In this case the free dimensions of the enclosure must be about 12 m, and the total dimensions must be about 14-15 m.

4. In measuring the noise generated by high-power transformers, /389
motors and generators it no longer makes any economical sense to build anechoic chambers, so that we must resort to improvisations in assemblying sheds and noise measurements must be conducted at periods during which the production process of the sheds is interrupted.

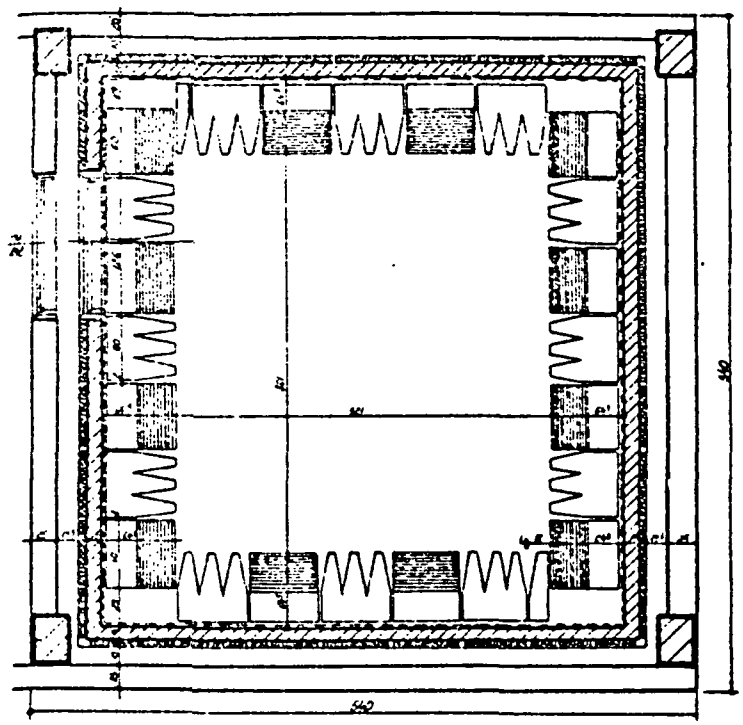

Fig. 2.

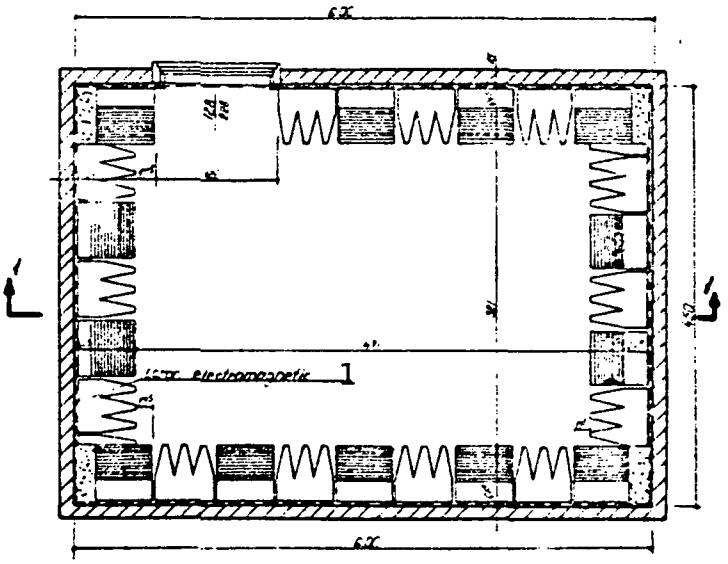

Fig. 3. Height plane ±0.00.
1. Electromagnetic screen.

All the anechoic chambers of the engineering industry are equipped with electromagnetic screens for the purpose of damping electromagnetic fields of about 60 dB within the 4-5 MHz frequency range.

It goes without saying that electric power is supplied to receivers mounted in screened anechoic chambers by means of filters so as to prevent perturbing waves from penetrating by way of the feed lines.

In the anechoic chambers mentioned in paragraph 1, the noise sources are suspended and the operating personnel circulates on nets placed above the floor of the enclosure.

In the anechoic chambers mentioned in paragraph 2, the equipment to be studied can no longer be suspended because of its great size and weight; in these cases the source fastening plate is so dimensioned that its proper vibration frequency will differ by one order of magnitude from that of the equipment to be studied.

Placing nets in small anechoic chambers constitutes a special problem, and in the event that this net is metallic it must be treated in a suitable manner with a view to avoiding perturbances of the electromagnetic field from the outside of the room toward the field of the net.

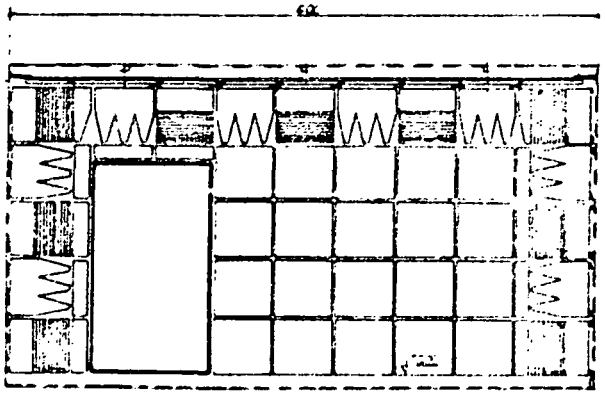

Fig. 4. Section 1.1.

The anechoic chambers described in paragraph 2 are suspended on springs, which prevents external vibrations from being transmitted to the inside of the room. In the case of larger anechoic chambers this solution becomes contraindicated from the economical point of view. The chamber will be placed at great distances from the vibration sources in quiet areas of the industrial shed. /390

In building anechoic chambers we used exclusively indigenous materials which gave complete satisfaction from the point of view of performance as well as behavior in time.

In the anechoic chamber of Fig. 2, the background noise has a value of about 25 dB, and in that of Fig. 3 the background noise is about 20 dB, both in the 2 kHz acoustic frequency range.

Summary

The level of noise generated by electrical engineering equipment constitutes one of the principal parameters that define the quality of the product. /392

As the general designer for electrical engineering and electronic plants, Electrouzinproiect in pursuit of improving the quality of products accorded particular attention to the checking installations for the noise level of electrical engineering products.

The paper describes some installations built by Electrouzinproiect for technological noise level checking installations.

Also given are some general considerations about the constructional elements and parameters of these installations.

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NA-28123

SOME ASPECTS OF THE BEHAVIOR OF MATERIALS USED IN MECHANICAL VIBRATION INSULATING TECHNIQUES

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Data about the physical mechanical properties of the materials /393 employed in the technology of vibration control must be as exact as possible in view of the many situations that arise in this field.

Insulating buildings from vibration sources by means of insulators is one of the principal problems of the technology of vibration control.

The rheological aspect of insulators is nicely represented by the elastic element, the damping element being of secondary importance. The principle on which these insulators work is the damping of the amplitudes of the vibrations transmitted to the building from the source by setting up an oscillating system with those materials which demonstrate sufficient elastic qualities to permit the oscillating system's proper frequency to drop far enough below the exciting frequency.

A necessity for the judicious design of insulators is knowledge of the relations between their static and dynamic rigidities.

In the present study we analyze the manner of variation of the ratio between the static and dynamic rigidities and, implicitly, the ratio between the corresponding moduli of elasticity, of some vibration insulating materials.

The dynamic modulus of elasticity of a material under a given axial stress is defined as the derivative of a function $\sigma(\varepsilon)$ in which σ denotes the effort resulting from the axial stress to which the material is subjected and ε denotes the specific elongation to which the effort σ corresponds.

By the static modulus of elasticity is meant the modulus of elasticity for the case when the curve of the function $\sigma(t)$ that indicates the variation of the stress as a function of time, is characterized by the existence of thresholds corresponding to the maintenance of an effort that is constant in time.

Deformations must be determined only at periods of time corresponding to constant values of the efforts, taking into account the values corresponding to the stabilization in time of the phenomenon.

In view of the foregoing the dynamic or static rigidity is determined as the dynamic or static modulus of elasticity referred to a given thickness of the material.

In the case of porous materials with open pores under the conditions of small tests the dynamic rigidity is the sum of the dynamic rigidity of the skeleton and the rigidity due to the elasticity of the air in the pores. As a result of periodic axial stresses a part of the air in the pores will be driven back or sucked in and the remaining air will contribute to the deformation strength of the material. /394

The materials analyzed in the course of the present study were as follows: mineral wadding, cellular polystyrene and Lombardy poplar agglomerates (ASKO and SUPEREX).

In general, in order to determine the dynamic rigidity and, implicitly, the dynamic modulus of elasticity resonance methods are employed.

The measuring instrument consists in principal of a mass-spring system in which the mass is made of the material that is to be tested. The mass of the instrument used in the present case was a concrete slab exerting a uniform load of 200 daN/m^2. For each material studied we determined the system's resonance frequency, and obtained the dynamic rigidity by means of the relation:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

in which: k is the dynamic rigidity and m is the mass of the system.

For the mineral wadding, which is the porous material with open pores, to the thusly determined dynamic rigidity we added the additional rigidity due to the deformation strength of the air remaining in the pores. This rigidity is determined by the air's flow resistance, defined as the ratio of the pressure difference between the faces of the sample (corresponding to the forced passage of air through the sample) and the volume of air passing per unit time per unit of sample surface.

The principal disadvantage of this method is the fact that it does not permit a complete picture to be obtained about the modulus as the dynamic rigidity differs for different load gradients, and we are left with a single value corresponding to the load that the mass of the system gives.

In principle, a sufficient number of different masses could be added to the research device in order be able to obtain a

satisfactory dynamic rigidity curve; from the practical point of view, however, the procedure would become unwieldy.

Then again, this method gives no indication about any variation that might occur in the dynamic rigidity (for a given load gradient) as a function of the speed with which this load was applied. Therefore, in order to obtain additional information about this problem, we studied axial compression in a Schoper press, applying the loads with a constant speed (speed of displacement of the dies for no-load operation = 80 mm/min).

We carried out the static studies for determination of the static rigidities and moduli of elasticity according to the principles set forth when we explained the notions of static modulus of elasticity and static rigidity, so that from the practical point of view we can consider the speed of applying the loads was zero.

In what follows we present in the form of tables the results obtained by these methods. /395

TABLE 1: DYNAMIC RIGIDITIES AND MODULI OF ELASTICITY DETERMINED BY THE RESONANCE METHOD (σ = 200 daN/m^2)

Characteristic	Sample thickness (cm)	Materials			
		Mineral wadding	Cellular polystyrene	ASKO	SUPEREX
Dynamic rigidity (daN/cm^3)	2.5	0.88	3.32		
	5	0.42	2.16		
	9				1
	12.5			1.75	
Dynamic modulus of elasticity (daN/cm^2)		2.3 - 3	8.3 - 10.8	21.9	9

Analysis of the experimental data obtained by these methods leads to the following conclusions:

1) Essentially, a material's modulus of elasticity depends on the shape of the $\sigma(t)$ curve, in which σ denotes the value of the stress and t denotes the time in which it increases from zero to the value under consideration.

By stressing a material in a field of high enough speeds we obtain a family of $\sigma(\varepsilon)$ curves that will have for their lower limit the curve resulting from the static study whose tangent to the origin will make a certain angle with the abscissa (see Figs. 1 and 2).

TABLE 2: STATIC AND DYNAMIC MODULI OF ELASTICITY (DETERMINED IN THE SCHOPER PRESS, THE SPEED OF DISPLACEMENT OF THE DIES FOR NO-LOAD OPERATION EQUALS 80 mm/min)

σ (daN/cm^2)	Moduli of elasticity E (daN/cm^2)			
	Mineral wadding	Cellular polystyrene	ASKO	SUPEREX
0,010	0,60 (1,11)x)	3,56 (4,70)	3,25	2,80
0,020	0,62 (1,16)	4,16 (5,25)	3,70	3,15
0,030	0,65 (1,20)	4,50 (7,00)	4,30	3,60
0,040	0,69 (1,22)	5,30 (7,27)	4,75	4,10
0,050	0,75 (1,25)	5,70 (7,70)	5,15	4,50

x) In parentheses are given the values of the moduli of elasticity determined in the Schoper press.

As the loading speed increases with constant values the $\sigma(\varepsilon)$ curves in the field of speeds under consideration will become increasingly dense. This is a consequence of the fact that the material's deformation speed becomes stable in proportion to the loading speed.

Every material can be characterized by such a family of curves that will have a larger or smaller scattering area according as the damping elements in its rheological circuit have a greater or smaller weight. For example, rubber, whose modulus of elasticity, practically speaking, does not depend on the loading speed, will have a family characterized by a single curve.

2) It can be seen from the curves of Figs. 3 and 4 that the static and dynamic moduli of elasticity vary essentially with the load gradient, so that a characterization by means of a single value of the dynamic modulus of elasticity-static modulus of elasticity ratio will not be sufficient. /396

In view of these situations we consider that for a sufficient characterization of a material's elastic qualities it is necessary to establish a family of $\sigma(\varepsilon)$ curves determined in a sufficiently varied field of speeds as a function of the different conditions under which it will have to be run. /397

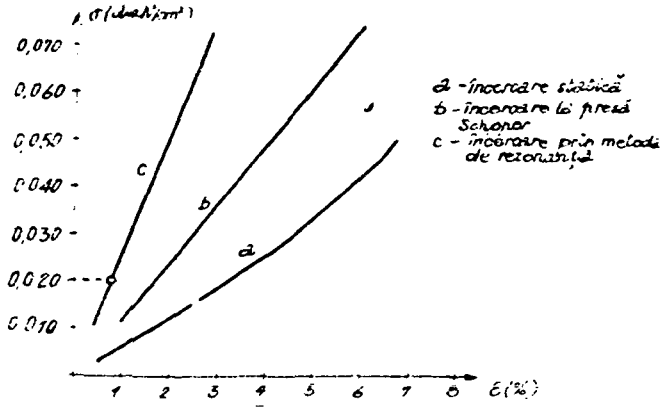

Fig. 1. σ - ε curves for mineral wadding. a) Static study; b) Schoper press study; c) Resonance method study.

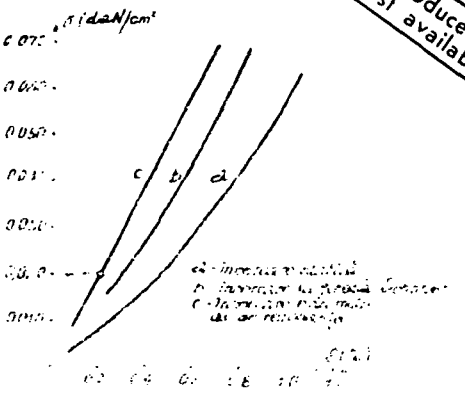

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Fig. 2. σ - ε curves for cellular polystyrene. a) Static study; b) Schoper press study; c) Resonance method study.

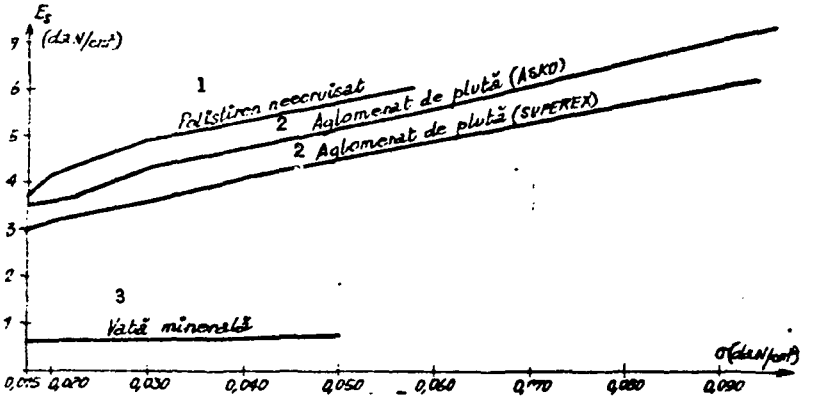

Fig. 3. Static moduli of elasticity versus load gradient. 1. Polystyrene (not cold-hardened); 2. Lombardy poplar agglomerate; 3. Mineral wadding.

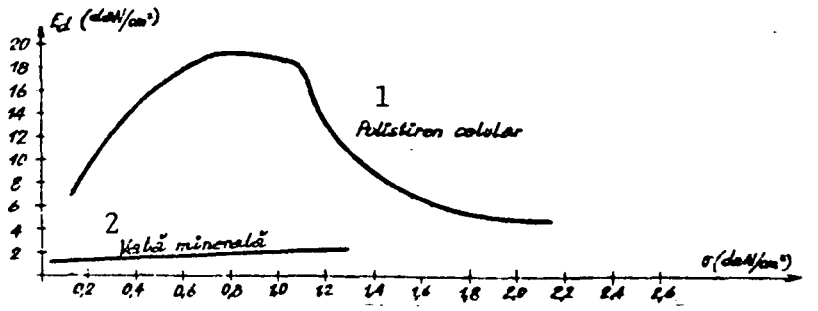

Fig. 4. Moduli of elasticity determined in Schoper press versus load gradient. 1. Cellular polystyrene; 2. Mineral wadding.

Summary

The paper brings to light the inadequacy of characterizing a vibration-insulating material by its dynamic modulus of elasticity determined by the resonance method. /398

It is shown that the dynamic modulus of elasticity actually depends on the material's loading speed as well as on the volume of the load, so that it is necessary to determine its variation as a function of the two parameters. The needed values are to be derived for actual conditions of use.

N74-28134

EFFICIENT SYSTEMS OF SOUND-INSULATING FLOORS

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Advances in the technology of house building necessitate a more rational use of traditional building materials from the engineering as well as the economic point of view. /399

According to the modern view, floors do not play only a finishing, decorative role, but constitute building elements, structures consisting of several layers, which complete the qualities of the floor that ensure comfort.

The soundproofing of floors refers to two types of noises: aerial noises and impact noises.

The principle of insulation against impact consists in desolidarization of the floors from the building's resistance skeleton for the purpose of insulating and localizing the noises that are transmitted through the floor.

Within the framework of a program of collaboration with production units we studied and tested efficient systems of sound-insulating floors.

1. Floor Structures

Our studies were aimed at determining the engineering and economic conditions for the utilization of a lamellar parquet, tempered fiber hardboards and tempered chipboards in building floor structures.

Our experiments in a residential building concerned the following types of floors:

-- a 10 mm thick lamellar parquet glued with adhesive based on APV (aracet EC) on the insulating fiberboard placed directly on equalizing sand or dry rubbish (Fig. 1);

-- a 10 mm thick lamellar parquet glued with adhesive based on APV (aracet EC) to a floating concrete slab (Fig. 2); / 400

-- 4 mm thick tempered fiberboard glued with adhesive based on APV to the insulating fiberboard (Fig. 3);

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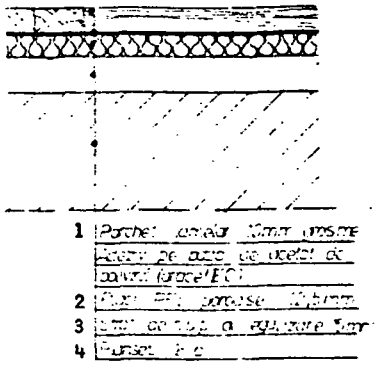

Fig. 1. 1. 10mm thick lamellar parquet adhesive based on polyvinyl acetate (arcet EC); 2. 12.5 mm thick porous fiberboards; 3. 15 mm thick layer of equalizing sand; 4. Reinforced concrete floor.

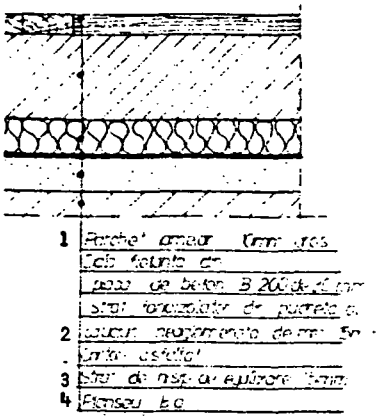

Fig. 2. 1. 10 mm thick lamellar parquet floating slab of: 30 mm thick plate of B 200 concrete; 2. 15 mm thick sound- insulating layer of unagglomerated rubber powder; roofing board; 3. 15 mm layer of equalizing sand; 4. Reinforced concrete floor.

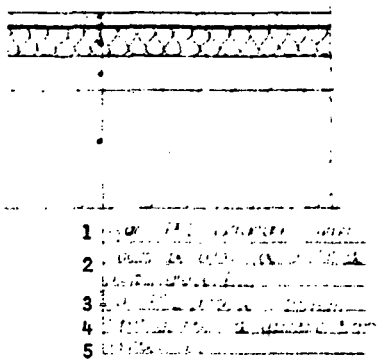

Fig. 3. 1. 4 mm thick tempered fiberboards; 2. Adhesive based on polyvinyl acetate (aracet EC); 3. 12.5 mm thick porous fiberboards; 4. 15 mm thick layer of equalizing sand; 5. Reinforced concrete floor.

-- 12 mm thick "PALEX" nailed to concrete fiberboards (Fig. 4).

Research and experiments on other types of floors such as:

-- a PVC carpet glued with APV adhesive on a 3.5 cm thick concrete slab (50 x 50 cm) placed directly on a layer of rubber powder;

-- a PVC carpet glued with APV adhesive on a 3.5 cm thick concrete slab placed on a layer of porous fiberwood.

2. Laying Method

The laying method is based on industrial procedures, which involve parquet panels, tempered fiberboard, PALEX and prefabricated floating slabs, without wetworking.

Lamellar parquet floors are layed as follows:

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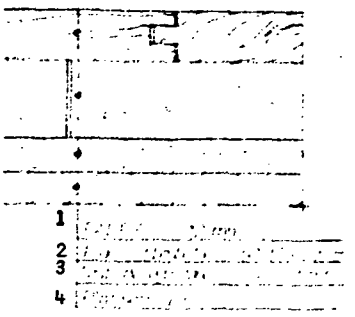

Fig. 4. 1. 12 mm thick PALEX; 2. 50 x 50 x 3.5 cm concrete fiberboards; 3. 15 mm thick equalizing sand; 4. Reinforced concrete floor.

-- insulating fiberboards are placed on a 15 mm thick layer of dry equalizing sand in such a manner that their sides form 45° angles with the sides of the lamellar parquet panels and with the walls of the room.

If a floating slab is utilized, over the layer of equalizing sand are placed a roofing board, a 15 mm thick sound-insulating layer of rubber powder, a B 200 concrete plate and the lamellar parquet panels. The parquet is scraped four days after gluing.

Tempered fiberboard and PALEX floors are layed as follows:

-- the equalizing sand is spread; porous fiberboard or concrete fiberboard are laid with their joints staggered with respect to the joints of the panels;

-- 10-24 hours before gluing, the tempered fiberboards or PALEX boards are moistened with water on the unfinished side; /401

-- the adhesive is applied to the surface of the support layer, and the final layer is applied by pressing.

For the purpose of industrializing the laying method we are creating compounded fiberboard floor slabs made of 5 mm thick tempered boards forming the wear board and 10 mm thick porous boards glued together and delivered in the formats 300 x 300; 400 x 400 and 500 x 500 mm.

The compounded fiberboard slabs are applied in one of the structural versions shown in Figs. 5 and 6.

Being colored in two shades, the slabs can be laid on the support in such a manner as to form various designs; placed with their joints woven or continuous, placed diagonally or parallel to the walls and placed in a checkerboard pattern or with boards in two columns (with braids). Sometimes tempered boards are equipped with parallel or diagonal trenches, which also have a decorative effect.

At the place where the walls intersect the floor wooden plinths are applied that cover the 5-10 mm joints between floor and wall, being screwed to the wall in wooden dowels.

Tempered fiberboard or PALEX board floors are cleaned or maintained in the following manner:

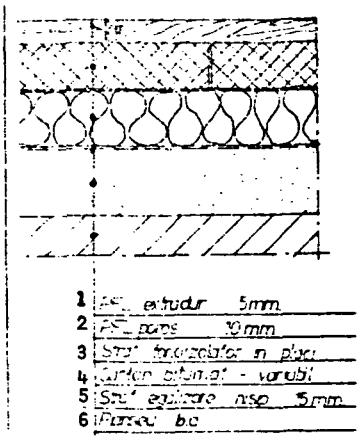

Fig. 5. 1. 5 mm thick tempered fiberboards; 2. 10mm thick porous fiberboards; 3. Sound-insulating layer in boards; 4. Bitumenized (variable) board; 5. 15 mm thick layer of equalizing sand; 6. Reinforced concrete floor.

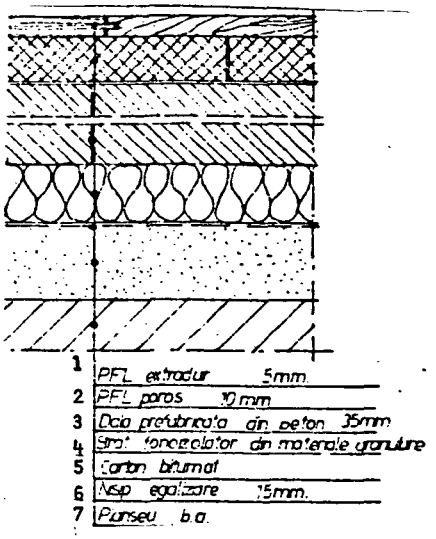
1 PFL extrudur 5mm
2 PFL poros 10 mm
3 Dala prefabricata din beton 35mm
4 Strat foneoizolator din materiale granulare
5 Carton bitumat
6 Nisip egalizare 15mm.
7 Planseu b.a.

Fig. 6. 1. 5 mm thick tempered fiberboards; 2. 10 mm thick porous fiberboards; 3. 35 mm thick prefabricated concrete slab; 4. Sound-insulating layer of granular materials; 5. Bitumenized board; 6. 15 mm thick layer of equalizing sand; 7. Reinforced concrete floor.

-- light stains are removed with wiping resin;

-- water-soluble stains are washed with soap and water;

-- grease stains are removed with detergents.

The final layer is covered with hard varnishes based on polyesters, applied in 2-3 layers, with a consumption of about 300 g/m^2 ready-made varnish (varnishes based on polyurethanes are recommended).

3. Sound-Insulating Capacity

By interposing an elastic, sound-insulating layer between rigid floor element and floor a reduction in the transmission of impact noise can be observed at frequencies higher than a resonance frequency given by the relation [4]:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k_d}{m_1}} \leq f_{adm}$$

where $k_d = \frac{E_d}{h}$ N/m^3 /402

where: k_d is the dynamic rigidity per unit surface of the elastic layer; E_d is the elastic material's dynamic modulus of elasticity, Table 1 [4].

TABLE I

Name of material	Dynamic modulus of elasticity E_d (N/cm^2)
Boards of rigid polystyrene foam	120
Rubber powder in bulk	100
Agglomerated Lombardy poplar boards	110
Porous fiberboards	160
Porcelain	25
Concrete fiberboards	70

The capacity for impact noise insulation was studied for the types of floors shown in Figs. 1, 2, 3, 4, 5 and 6, and the results are shown in the form of impact noise insulation curves (Fig. 7) and estimated by comparing them with the STAS 6156-60/68 limiting curve. Examination of the curves plotted on the basis of our research leads to the following conclusions:

-- all the structures in the floating slab system yielded good results at the limit of the normalized impact level;

-- curve 2 -- the floor structure shown in Fig. 1 -- exceeds the normalized curve in the 60-400 Hz frequency range by 4 dB and shows a limiting damping below a frequency of 60 Hz and above 400 Hz;

-- curve 3 -- the floor structure shown in Fig. 2 -- exceeds /403
the normalized curve in the 100-400 Hz frequency range by 2 dB and shows better impact noise damping than the solution of Fig. 1;

-- curve 4 -- the floor structure shown in Fig. 6 -- yields measurement results that fall below STAS limiting curve (1) and behaves best throughout the entire frequency range;

-- curve 5 -- the floor structure shown in Fig. 3 -- exceeds by at most 2 dB the STAS curve in the 180-380 Hz frequency range and for the rest falls below the STAS limiting curve, which is better behavior than the solutions of Figs. 1 and 2;

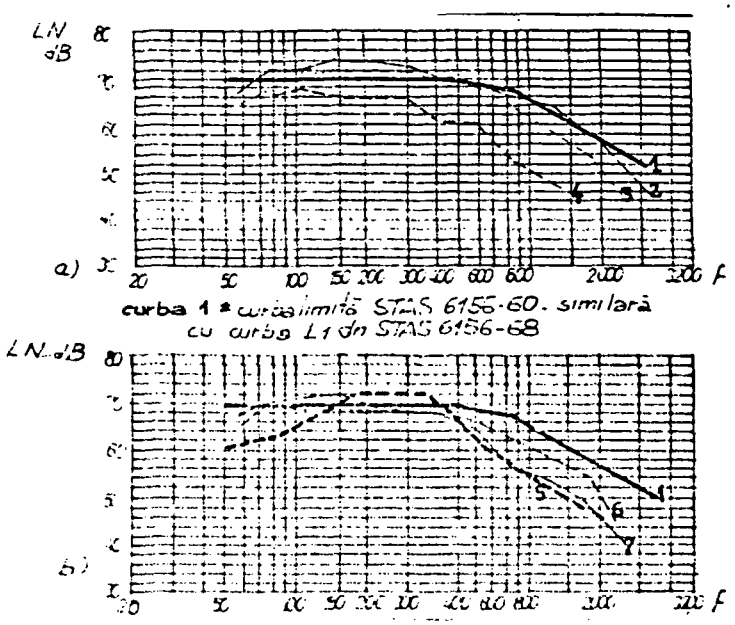

Fig. 7. 1. Curve 1 = STAS 6156-60 delimiting curve, similar to the STAS 6156-68 L1 dn curve.

-- curve 6 -- the floor structure of Fig. 4 -- exceeds the STAS curve by 2 dB in the 70-200 Hz frequency range (practically unappreciable for floors of category II-a) and falls below it for the rest;

-- curve 7 -- the floor structure of Fig. 5 -- falls below the STAS limiting curve but shows an almost limiting behavior for the 60-400 Hz frequency range.

Conclusions

The purpose of our studies and research was to establish efficient solutions for floors made of superior wood products.

The engineering and economic efficiency of the solutions presented as recommendable versions (Figs. 3, 4, 6) were established in comparison with traditional solutions with a view to the criterion of cost per unit of acoustic comfort brought about.

On the basis of the curves plotted in Fig. 7 the type of floor structure can be chosen as a function of the impact noise level admissible for the building.

Our research is part of the national noise and vibration control program for building.

Summary

The paper summarizes the results of experiments on dwelling houses with a view to the creation of efficient warm floors. /406

The experiments concern the types of floors built in Rumania with currently produced materials. At the present time experiments are continuing with solutions based on concrete slabs equipped with a PVC carpet and placed on an elastic layer.

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N74-28125

CONTRIBUTIONS TO THE ACOUSTIC PROTECTION OF CIVIL BUILDINGS TAKING INTO ACCOUNT THE USE OF LIGHT PREFABS

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Although in Rumania the industrialization of building by using /407
light prefabricated elements has not become generalized, what has been accomplished in this direction isever more significant and evolutionary tendencies are very clear. This is because the possibility of increasing productivity, eliminating wet procedures, the elasticity of the architectural element and the light weight of the buildings in the aggregate have an effect on the relatively high cost.

But, as is well-known, reducing the weight of building elements contradicts the law of mass from the point of view of sound-insulating qualities. It is also known that using light materials in order to obtain increased acoustic damping coefficients leads to complex structures and, implicitly, to increased cost price.

It is, therefore, necessary to approach the problems of the acoustic protection of buildings in parallel with the designing, testing and introducing of new types of elements with due regard also for the great diversity of functions to which they will be subjected.

So vast is the problem that the present paper will refer only to the outer and separating walls used in social-administrative buildings and residences.

Today, heavy prefabricated outer walls for civil buildings are in current use in Rumania. But making their presence ever more felt in social-administrative buildings and, more rarely, residences throughout the world are "curtain walls". These are prefabricated outer walls, usually continuous with the front of the structure, with glassed and opaque parts, and having complex structures. Their principal characteristics are: an increase in glassed surfaces, reduced thickness (5-10 cm) and the use of light materials.

Indications concerning the conditions of acoustic insulation of outer walls usually have the character of recommendations. This situation is especially due to the presence of windows which, as we shall see, radically change the qualities of the facade as a whole. It is possible, however, to formulate some requirements /408

that must be satisfied with respect to the damping of the transmission of noises from the outside, more rarely the damping of the transmission of noises from the inside, the prevention of the birth or transmission of structural noises and the avoidance of acoustic bridges between the curtain walls and the inner structure of walls and ceilings.

As far as insulation against aerial noises is concerned, by superposing the opaque parts in layers it is usually possible to obtain damping coefficients R'_m of about 30-35 dB for a weight of 30-40 kg/m^2, which at first blush seems rather little.

But the degree of insulation of the wall is determined on the whole by the weakest points, namely, the windows. For this reason, and taking into account the present tendency to increase the proportion of glassed surfaces, in comparing two similar facades, one with a heavy wall and the other with light prefabricated panels, there will exist practically no difference in sound insulation. If we take the example of a 10 m^2 heavy face wall with R'_m = 47 dB and having a 3 m^2 window (R' = 25 dB), the mean damping coefficients of the facade on the whole will be about 30 dB. Such a coefficient can also be easily attained with a light wall by paying particular attention to the windows (by increasing their air tightness and thickening the glass) and thus bringing them up to a coefficient R' of 30 dB, similar to that of the opaque parts of the curtain wall. It also ensues that it would be useless to make additional efforts, which would, of course, also lead to a higher cost, to increase the damping coefficient of the solid parts. On the other hand, care must be taken to close all connecting joints and orifices that appear in windows, opaque panels, etc., because, as is known, even the smallest openings can cancel practically all expended efforts.

As far as our structural noises are concerned, they can be received directly by the facade (rain and hail on the outside, and tenants on the inside), but they can even arise in its elements as a result of thermal shocks in case the skeleton of the facade or its grippers resist temperature-induced deformations.

Structural noise control depends on a supple construction of the skeleton by means of suitably tightened deformation joints and by attaching the resistance structure with the aid of elastic elements.

The most delicate problem in the acoustic protection of curtain walls is the acoustic bridges that arise in the joints between the continuous facades and the resistance structure, which permit noise transmission between adjoining rooms. These joints serve the dimensional tolerances of the construction and the placement of anchorage devices. The tamping of these joints with elastic materials,

special profiles, or, in some cases, even with concrete (which solution, however, renders deformations difficult), must be kept in mind as early as the designing stage and must be executed with care at the building site.

While, as we have seen, windows considered to be closed constitute the most delicate points, when they are opened for ventilation purposes, the facades lose practically all of their soundproofing qualities. Therefore, the correct approach to the problem of light facades must begin with the placement and the general architectural plan, first of all moving out of the way rooms that must be protected from surrounding noises. /409

The only radical solution from this point of view would be complete tightening of the facade, accompanied by an air conditioning plant on the inside. Unfortunately, this solution is difficult to implement because of its very high cost.

While, as we have seen above, with light outer walls a degree of soundproofing can be achieved that is equivalent to that obtained in the case of heavy facades, in the case of separating walls the problem is more difficult. On the one hand, more complex problems of functionality crop up and, on the other, soundproofing demands are more categorical.

The first point to be made is that light, prefabricated, separating walls are -- at least for the time being -- more expensive than the classical solutions, even if the problem of soundproofing does not come up at all. Using them is first of all a response to the demands of productivity, industrialization and facilitation of construction.

Under these conditions it is important that the soundproofing of walls, which usually entails an additional cost, be done with discretion without, however, neglecting the problem of acoustic protection.

As regards the problem of the damping coefficients of the walls of civil buildings as a function of degrees of comfort (STAS-6156-68), this too may have to be tackled in a more subtle manner by soundproofing other walls where this may be useful, not using insulating walls where their qualities would not be completely utilized and, finally, taking into account the contribution of the general acoustic protection measures involving architectural elements and making suitable use of the possibilities offered by acoustic absorption.

In office buildings, the separating walls between rooms or "functional units" can be treated differently depending on the nature of the work and, especially, the number of persons quartered.

In rooms with more than 10-15 persons, which are being used ever more frequently, the noise due to current activity assumes a continuous character and rather high levels. The situation is the same where typewriters exist. Under such conditions any noises arriving from the outside are less disturbing, and wall soundproofing requirements are fewer than in the case of rooms for few persons. In such rooms, however, greater importance is assumed by soundproofing by means of adequate absorption. /410

In order to meet such varied demands we propose the creation of standardized (and therefore, amenable to machine production), self-supporting wall panels with an asymetric internal structure and having a reflecting surface and a partially or totally absorbing surface. They would combine a space delimiting function with an additional absorbing function and could be built singly, doubly or even in three layers, whereby they would constitute walls with varied insulating and absorbing qualities.

It would have to be possible to dismount the panels in order to permit subsequent changes in plan or corrections at those points where sound insulation turns out to be insufficient and, finally, in order to be able to mount them above the floor so as to facilitate the continuity of any floating slab that might be used.

Figure 1 shows some typical wall structures as a function of rooms that are separated.

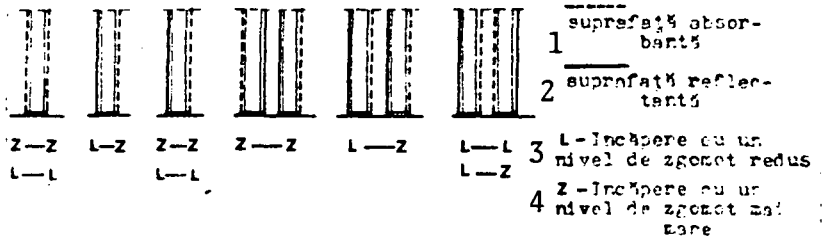


Fig. 1. Diagram of separating walls of light panels with reflecting and absorbing surfaces. Cases 1-2, 3-4, 5-6 have increasing damping coefficients. Cases 1 and 2 presuppose walls with doors. 1. Absorbing surface; 2. Reflecting surface; 3. Room with reduced noise level; 4. Room with high noise level.

Here too, as in the case of outer walls, account must be taken of possible gaps in the walls, which radically change the wall's mean damping coefficients. For example, in the case of a 12 m^2 wall having R' = 30 dB and a 2 m^2 door (R'_m = 20 dB) we get on the whole a mean damping coefficient of 26 dB. If the walls had R' = 44 dB (at almost twice the cost), the mean damping coefficients would increase by only 2 dB. But we arrive at the same result by adding 4-5 dB to the damping coefficient of the door. In such

cases it would be preferable to use the sound-absorbing surfaces of the panels. An additional mean absorption coefficient of 0.6 on a 12 m^2 surface can lead in a 60 m^3 room to a drop in the reverberation time by 0.5 sec and a drop in the sound level in the room by 3-5 dB, with a gain in quietness for both rooms. /411

The present tendency is for office buildings to have bigger offices with a smaller number of separating walls ("landscape" type). Taking into account more judicious utilization of surfaces and elimination of separating walls, solutions of this type are more economical in spite of the cost of the sound-absorbing treatment that in such cases becomes obligatory.

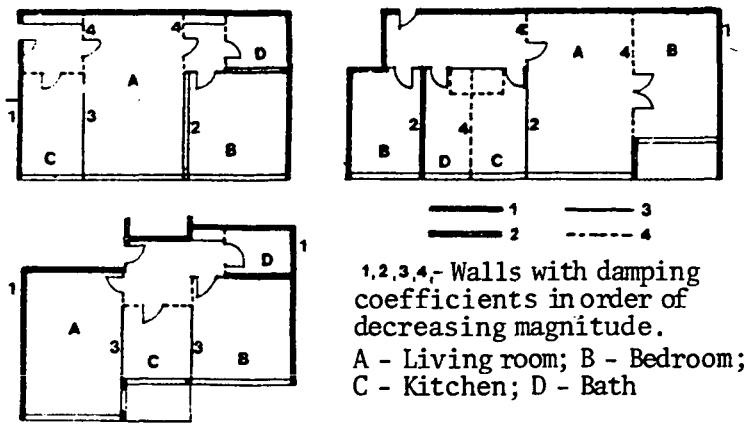
In rooms with few persons, where there is no proper background noise, an additional element would have to be taken into account: the masking effect of the background noise from the outside. With a background noise of 45 dB, a 70 dB conversation in a neighboring room penetrating through an interior wall with R' = 35 dB, does not disturb activity. If, however, the background noise in the inside drops to, say, 30 dB owing to a quieter location, the same separating wall no longer suffices.

In continuing our discussions of the implications of using light separating walls in building residences, the walls of apartment interiors must be approached separately from the others.

At the present time, in the interior heavy walls are used which are made of brick (33 dB) or prefabricated from cellular concrete (d5 dB). The transition to walls made of prefabricated panels with simple and light structures having R'_m around 30 dB does not raise special problems of acoustical protection because the rooms of an apartment usually communicate with one another. Using such walls, which can even be immobile, is recommended in view of the present tendency toward flexible apartment structures.

On the basis of more careful study interior walls of apartments can also be hierarchized in terms of damping coefficients (and therefore, price) as a function of the category to which they belong (Fig. 2). To the first category would belong walls between living rooms (2); to the second category would belong walls between living rooms and annexes(3); and to the third category would belong walls in which there are doors, and which can have a damping coefficient close to that of doors, properly speaking (4).

How to delimit apartment walls depends on the resistance structure chosen. By now, in Rumania, where we use almost exclusively diaphragm-walls made of brick or monolithic built-up reinforced concrete ($R'_m \cong 49$ dB), the problem of sound insulation has been solved (1 -- Fig. 2).



1,2,3,4,- Walls with damping coefficients in order of decreasing magnitude.
A - Living room; B - Bedroom; C - Kitchen; D - Bath

Fig. 2. How to differentiate interior walls of residences on the basis of damping coefficients.

In the case of structures with a skeleton we can use heavy, double, prefabricated walls or, possibly, in combination with light materials. The utilization of light walls will be justified only in the case of buildings with many stories (more than 18-20) because the R' curve demanded by STAS 6156-68 is rather hard to obtain. /411

Conclusions

-- The careful use of light prefabricated walls of facades does not generally raise special problems of acoustic protection because the acoustic insulation of the architectural complex of facades is comparable with that of classical solutions with heavy walls.

-- Necessary, on the other hand, are experimental studies of the behavior of interior wall panels with an asymmetric structure and asymmetric reflecting and absorbing surfaces, taking into account their position with respect to the noise source.

-- Because wall panels based on plates made of wood or fiber chips do not meet NPCI specifications, it is necessary to experiment with wall structures based an asbestos cement, mineral wadding, gypsum board and plastics.

-- In order to reduce the price of buildings in which light walls are employed (cheaper than the usual heavy walls), it is necessary to study carefully the architectural elements with the following aims:

-- maximum reduction in the number of compartments;

-- grouping and separation of functions in terms of noise level;

-- the building, on the basis of modulated and standardized panels, of walls with characteristics that vary with the particular case.

Summary

Usually it is expensive to obtain suitable damping coefficients for exterior and interior walls made up of light materials. Hence, acoustic protection problems must be tackled in parallel with the design and introduction of new types of elements. /412

On the basis of a comparison with classical solutions, the author arrives at the conclusion that interior separating walls raise the most difficult problems.

In order to reduce building costs it is necessary to make a careful study of architectural solutions and to provide for wall structures with characteristics that vary with the particular case.

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CONCERNING THE DAMPING COEFFICIENTS OF EXTERIOR BUILDING ELEMENTS FOR AERIAL NOISE

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1. General Considerations

Owing to the development of industries and, especially, means of conveyance, the exterior noise level has grown considerably. /413

New buildings with their modern structures have characteristics that are conducive to noise transmission owing to the continuity of their structural elements and the fact that their exterior walls have a low mass as compared with traditional structures. Of interest under these conditions are knowledge of the variation in time of the noise level and more rigorous determination of the insulating characteristics of exterior walls. For this purpose we carried out measurements of the noise level in a building with P + 1 stories, having a bearing masonry structure, located on a street with relatively light traffic; we also recorded simultaneously the exterior and interior noise, as well as the intensity of the traffic on the street in question and that of a street with heavy traffic.

2. Experimental Characteristics of Exterior and Interior Noise on a Street with Light Traffic

The total noise levels due to urban means of conveyance vary with the number of vehicles that pass per unit time (one hour). According to the traffic data of Fig. 1, the number of vehicles in the street with light traffic varies but little in a day, ranging from 157 (minimum) to 265 (maximum), with a uniformity coefficient equal to 0.6, unlike the street with heavy traffic, where the number of vehicles ranges from 187 to 552, the uniformity coefficient being 0.34. /414

The statistical interpretation of the sound intensity level as a function of time is shown in Fig. 2. It can be seen that outside the building the maximum duration (20.8%) of the sound intensity level corresponds to a noise of 50 dB_A; noises of 85...90 dB_A have a very short duration: 0.07%.

Inside the building, the maximum duration (26.2%) corresponds to a noise of 35 dB, the maximum noise (70...75 dB) having a duration of 0.01%.

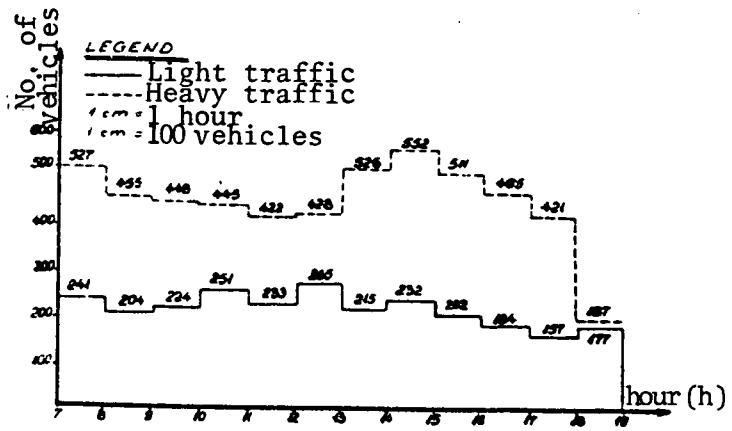

Fig. 1. Variation in the number of vehicles per hour in two streets with different traffic loads.

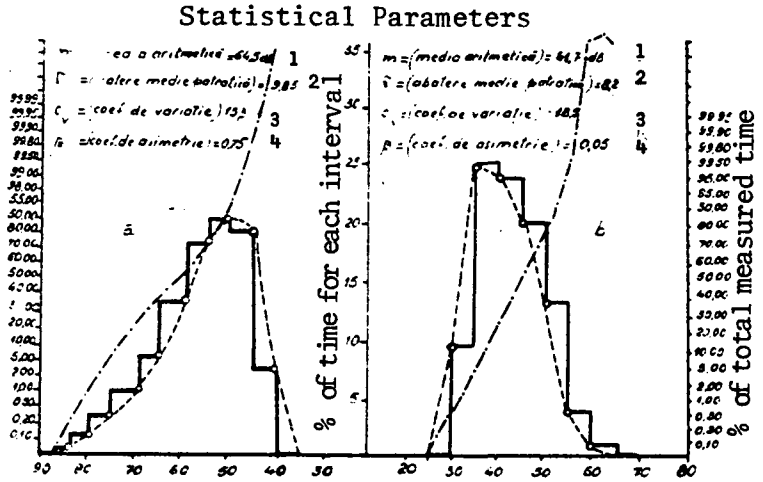

Fig. 2. Relative distribution of sound intensity level in time: a) exterior; b) room interior. 1. Arithmetic mean; 2. Mean square deviation; 3. Coefficient of variation; 4. Coefficient of asymmetry.

3. Estimation of the Degree of Sound Insulation of the Face Wall on the Basis of Statistical Interpretation of the Noise Level

In order to elucidate /415 the acoustic characteristics of constructions, analytic and experimental methods are used.

The current analytic methods, which are based on stationary methods, used in estimating the acoustic capacity of building elements do not characterize their complex response to the transmission of vibration noises. In practice, in order to determine the damping coefficients of building elements some empirical relations are utilized that are based on the experimental law of mass. In the case under study, taking into consideration the surfaces of the solid wall and of the joinery, we determined the sound reduction coefficients R_t in the usual frequency bands with the relation:

$$R_t = R_1 - 10 \log \frac{\frac{S_1}{S_2} + 10^{\frac{R_1 - R_2}{10}}}{1 + \frac{S_1}{S_2}} \qquad (1)$$

The values obtained /416 are shown in Table 1.

The experimental methods recommended by present standards envisage determination of the sound reduction coefficient R by measuring the intensity level of the noise outside the room L_1

TABLE 1

Frequency (Hz)	100	200	400	800	1600	3200
Sound reduction coefficients (dB_A)	25	29	33	37	41	45

and inside the room L_2, to which must also be added the influence of absorption in the receiving room according to the relation:

$$R = L_1 - L_2 + 10 \log \frac{A_0}{A} \qquad (2)$$

This method presupposes the use of stationary noise sources, which do not express the random character of outside noise inasmuch as it is known that outside noises can also be characterized from the energy point of view by their acting time.

Taking into consideration the continuous change in the outside sound field, some specialists reduced the problem to its energy aspect, proposing as the value of the outside noise level, the equivalent intensity level according to the relation:

$$L_{ech} = \frac{q}{0,3} \lg \left(\frac{1}{T} \sum_{i=1}^{n} 10^{\frac{0,3\, L_i}{q}} \cdot t_i \right) \qquad (3)$$

In order to estimate the acoustic insulation coefficients of building elements for outside noise, STAS 6156/68 fixes the maximum admissible values of the inside equivalent noise. Calculating the values of the equivalent level of outside noise (L_{ech} = 70 dB_A) and inside noise (L_{ech} = 40 dB_A) we propose that the degree of insulation of the exterior wall be estimated with the relation: /417

$$R_{ech} = L_{ech_e} - L_{ech_i} \qquad (4)$$

In the case under study the result was R_{ech} = 30 dB_A.

The inside equivalent level of 40 dB_A is higher than the admissible values of 30 dB_A, although the building is located on a street with relatively light traffic.

In order to elucidate the acting time of the noise we determined the levels L_{10}, L_{50}, L_{90} outside and inside the investigated

room. Figs. 3, 4 and 5 show the frequency variation of these levels.

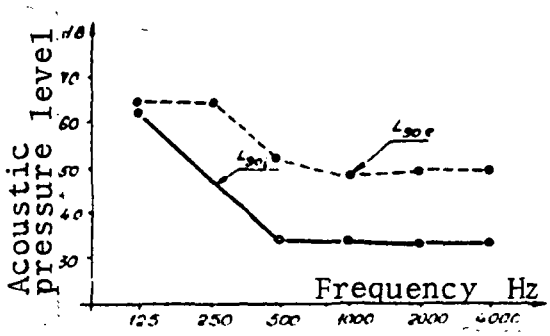

Fig. 3. Variation of outside and inside L_{90} as a function of frequency.

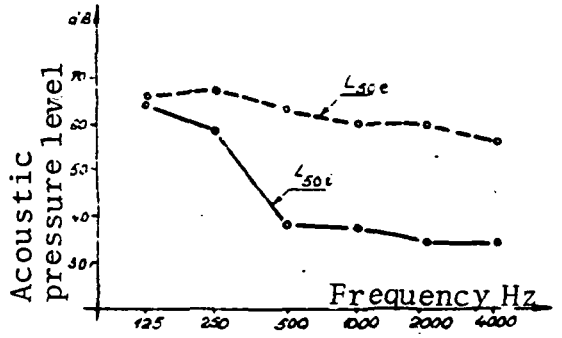

Fig. 4. Variation of outside and inside L_{50} as a function of frequency.

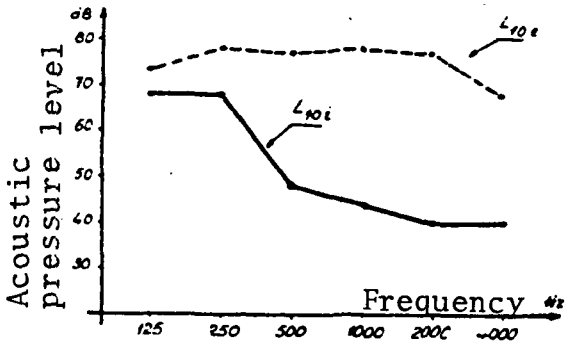

Fig. 5. Variation of outside and inside L_{10} as a function of frequency.

A general characteristic of all /418
the above-mentioned levels measured inside the room is the presence of higher values in the low frequency range (60...70 dB_A) and lower values in the intermediate (40...50 dB_A) and high (35...40 dB_A) frequency range. So it follows that the building element (the walls) are /419
less effective at screening low frequencies.

It can also be observed that the difference between the outside and inside noise levels increases as the duration of the noise decreases.

This might be due to the fact that a short duration of noise does not succeed in causing the building element to vibrate.

Assuming that these differences in the levels L_{10}, L_{50}, L_{90} measured outside and inside the room represent the sound reduction coefficients R_{10}, R_{50}, R_{90}, Fig. 6 shows their variation with the frequency.

Inasmuch as the level L_{90} has a rather long acting time as compared with that of outside noise we might regard it as a stationary noise.

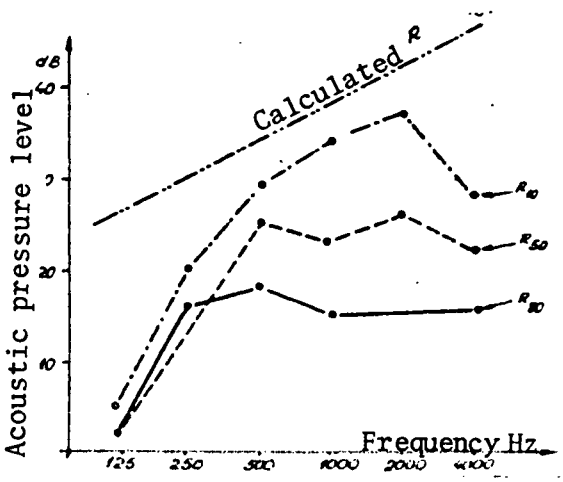

Fig. 6. Variation of sound reduction coefficients R_{90}, R_{50}, R_{10}, R calculated as a function of frequency.

In this case the values of the reduction coefficient R_{90} (15 dB_A) could be compared with those of the analytically determined coefficients (about 35 dB_A). /420

It follows from Fig. 6 that sound reduction coefficient R_{90} is much lower than that calculated on the basis of mass, which does not take into account the proper vibration frequencies of the joinery, whose surface is 1/4 of the total area and influences decisively the element's sound reducing and damping capacity.

If we take into consideration the mean value of R_{90} and the difference between the outside and inside levels for the maximum duration we get a value on the same order of magnitude (15 dB) that we can regard as a general characteristic of the element's sound reduction.

Summary

The values employed to define the sound reduction coefficient R of exterior walls for aerial noise do not take into account the continuous increase in the outside noise level (especially in the case of traffic noise) nor its energy variation as a function of time. /421

The authors of the present paper analyze these aspects by estimating the outside and inside noise levels on the basis of calculating an equivalent noise level (L_{ech}), a level with a maximum duration and the values of L_{10}, L_{50} and L_{90}.

The values obtained permit determination of the wall's sound reduction coefficients that are compared with those obtained by the customary method. It was found that the sound reduction values of the wall coincide for the case of sound levels with a maximum duration and the values of L_{90}.

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TRANSPARENT SOUND SCREENS

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In the practice of noise control situations are known in which /423
the presence of massive sound-insulating walls between noise sources and the zone that must be protected is not to be recommended.

For such cases we have worked out a type of modulated sound screen; by giving them the desired outline noise sources of any shape can be screened, creating zones with a lower noise level. Since they are built of transparent material (glass or styplex) they do not obstruct the supervision of installations and in the case of use in administrative buildings, computing centers, design bureaus, etc., they eliminate the appearance of claustrophobia, which is frequently caused by the fragmentation of rooms by means of opaque screens.

Solution Adopted

Unlike sound-insulating walls, sound screens made of materials /424
with a low sound absorption coefficient only retain a great part of the sound energy transmitted directly by the source but let pass with negligible damping reverberated components and components refracted by their upper edges. As a result, in the protected zone three categories of noise components will appear (Fig. 3):

(1) the transmitted noise;

(2) components refracted by the upper edge, possibly also by the free lateral edges;

(3) components reverberated by the ceiling or other reflecting surfaces.

In what follows, these components will be treated separately.

1. Transmitted noise. In principle, the damping introduced by a screen in the way of a direct wave is subject to the law of masses

$$TL = k.\log.M_s + r \quad /dB/$$

It follows that the sound screen must be built of material with a high enough specific mass. A 5 mm thick pane of glass provides a transmission damping of about 26 dB.

2. Refracted components have more significance in low-frequency bands. The refracted fraction was estimated on the basis of the angle of refraction Ø referred to the screen's virtual height h (see Fig. 1). With the aid of the nomogram of Fig. 2 the combined effect of angle Ø and of the virtual height h can be estimated for different frequency bands. In view of the fact that the

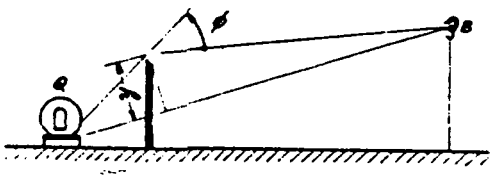

Fig. 1.

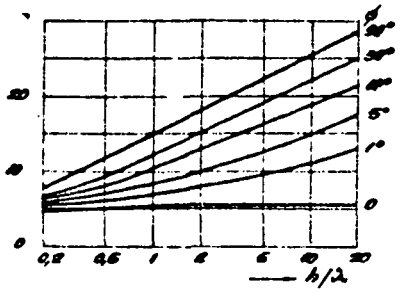

Fig. 2.

effect of the refracted wave is important only for low frequencies, sound screens provide efficient enough acoustic protection for intermediate and high sounds. For example, in the model of the sound screen we built, the attenuation was 16 dB for a frequency of 250 Hz, attaining 24 dB for 2,000 Hz. /425

2. Reverberated waves can bring the greatest fraction of acoustic energy to the protected zone. In order to reduce this effect the ceiling of the room and a part of the walls must be covered with sound-absorbing structures. Under such conditions, the reverberated component drops to an acceptable value, on the same order of magnitude as that of the refracted component.

Summation of components 1-3 (see Fig. 3) shows that under the above-mentioned conditions the protected zone can benefit from a reduction in the acoustic pressure level on the order of 12-15 dB in the 125-250 Hz frequency bands and of 18 dB in the higher frequency range.

This order of magnitude is sufficient in many cases and proves to be particularly useful in the case of these sound screens.

Such transparent sound screens are now being tested in a turbocompressor room (Fig. 4). The total noise level in the room exceeds 100 dB, in which high-frequency components predominate, and sound-absorbing treatment applied to the ceiling and a part of the walls does not improve the noise situation by more than 6-8 dB. In terms of frequency bands the situation is as follows (see table): /426

Frequency band	Hz	125	250	500	1000	2000	4000
Initial situation (dB)		85	94	95	91	92	86
Situation after application of sound-absorbing treatment	Estimated	83	88	89	83	84	79
	Realized	83	84	85	82	84	80
Situation after mounting sound screens	Estimate	80	85	84	78	76	69
Minimum damping due to sound screens		3	-	1	4	8	11

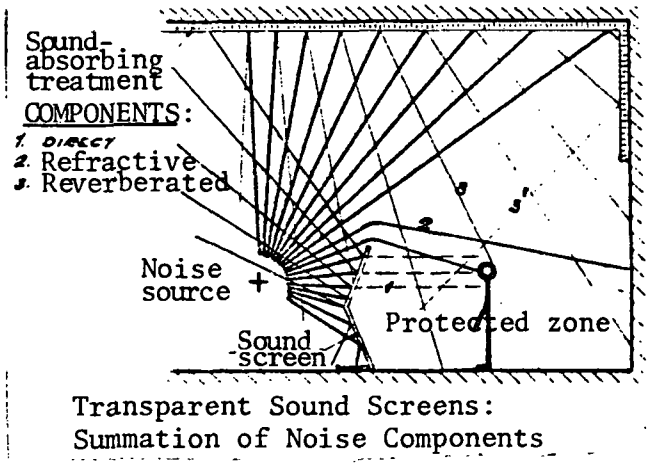

Fig. 3.

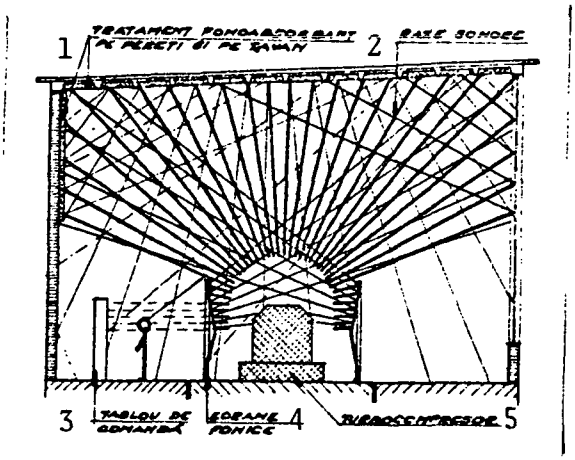

Fig. 4. 1. Sound-absorbing treatment of walls and ceiling; 2. Sound radii; 3. Control panel; 4. Sound screens; 5. Turbocompressor.

Sound screens are also being tested in a computing center.

It can be concluded that, in addition to improving the noise situation at rational costs, transparent sound screens will become commonplace because of their ergonomic advantages as compared with opaque sound screens. /427

Summary

The present paper describes transparent sound screens consisting of glass (organic glass) plates placed at fixed angles with respect to the floor. /428

Noise levels in the screened area depend on the cumulative effect of direct, reverberated and refracted components of sound energy radiated by the source. This effect is analyzed on the analogy of the summation of electrical impedances.

Inasmuch as under given circumstances sufficient noise control can be obtained, transparent screens seem to solve the problem of unimpeded supervision of installations in noisy workshops and power plants; in administrative buildings, computing centers, design bureaus, etc., they ensure acoustic comfort without space losses and without inspiring claustrophobia.

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N74-28125

CONCERNING THE SOUND INSULATION OF BUILDING ELEMENTS MADE UP OF LIGHT CONCRETES

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School of Architecture

In the building industry light concretes are used as walls for /429
insulating rooms against aerial noises and in the case of floors, against impact noises.

The light aggregates utilized in Rumania are: expanded clay (granulite) and expanded slag.

Our research concerns granulite concretes and expanded slag concretes [4].

Building elements made up of light concretes can also be employed as sound-absorbing elements for the purpose of reducing reverberation time, damping noises in rooms with noise sources or rooms that require special acoustic conditions.

The sound-insulating capacity is determined as a function of frequency in the 100-3,200 Hz band with the following relations [4, 6]:

Aerial noise:

$$R = 20 \lg (k_1 \cdot i_p m) + 20 \lg k_2 f - 28 \geq R_{\text{min. adm.}} \qquad (1)$$

Impact noise:

$$L_N = 20 \lg \frac{k_2 f^{\frac{1}{4}}}{E^{\frac{2}{8}} \cdot \rho_a^{\frac{5}{8}} \cdot d^{\frac{4}{7}}} \leq L_{N\,\text{adm.}} \qquad (2)$$

where:

k_1 is the correction coefficient for the light concrete structure;

m is the weight of the building element made up of light concrete (kg/m^2);

f is the frequency of the sound source, in Hz; for the 100- /430
3,200 Hz frequency band we work with the mean frequency:

$$f_{med} = \sqrt{100 \times 3200} = 565 \text{ Hz.}$$

E is the modulus of elasticity of light concrete (g/cm^2);

ρ_a is the apparent density of light concrete (g/cm^3);

d is the thickness of the building element (cm);

i_p is the coefficient of diminution of the sound insulation of the mass of the element under the effect of open pores;

$k_2 = 1/k_1$.

The purpose of the formulas proposed for calculation of the sound-insulating capacity for aerial and impact noises is to elucidate the physical characteristics of light concrete in order to establish the most exact correlation possible between calculated R and experimental R from measurements in experimental studies.

1. Aerial Noise Insulation

Investigation of the aerial noise insulation of light concrete building elements did not lead to unanimously accepted conclusions. Some researchers tried to advance new theories in calculating the sound-insulating capacity of light concrete while others tried to particularize the "law of mass" by means of the specific structural characteristics of light concrete. The present study starts from the latter idea and by comparing heavy concrete with light concrete I interpret the results of some applied research conducted during experimental studies.

In the case of two walls -- one of heavy concrete and the other of light concrete -- of the same weight m (kg/m^2), subjected to aerial noises of the same frequency f (Hz), according to the law of masses the reduction coefficients R_{calc} for aerial noise are equal. Measurements in the case of granulite concretes (GC) as well as expanded slag concretes (SC) disclosed that the reduction coefficient R_{exp} for aerial noise is greater than R_{calc} calculated according to the law of mass. The difference is considerable (+12-+6 dB) in the case of inferior brands (B50; B75; B100; B150; B200) and becomes practically admissible (+4-+2 dB) as the brand improves (B250; B300; B400).

On the basis of numerous tests and measurements I arrived at /431
the conclusion that the single factor that favorably influences the value of R_{exp} in respect of R_{calc} is the ratio of the mass of the light aggregate (m'_1, kg/m^3) to the mass of the cement stone (sand + cement + water, m'_2 in kg/m^3).

The coefficient $k_1 = m'_1/m'_2$ characterizes the structure of light concrete, i.e., the ratio of the proper frequencies for the light aggregate and the cement stone, as well as the bonds (elastic, semi-rigid and rigid) between the grains of the light aggregate, which defines the damping of internal vibrations by the bonds.

Table 1 shows the coefficients k_1.

TABLE 1

Brand of concrete	m'_1		m'_2		k_1	
	GC	SC	GC	SC	GC	SC
B50	900	1100	530	485	1,55	2,25
B75	1000	1200	660	580	1,53	2,06
B100	1020	1250	700	630	1,46	1,98
B150	1030	1250	720	675	1,43	1,85
B200	1045	1250	725	720	1,43	1,74
B250	1055	-	745	-	1,41	-
B300	1070	-	780	-	1,37	-
B400	1000	-	900	-	1,12	-

The much greater coefficients k_i in the case of expanded slag concretes as compared with those for granulite concretes are due to the difference in the apparent density for the weight.

In the case of light concrete building elements, the absorbed acoustic energy is transformed into thermal energy. This transformation always occurs as a result of friction and irreversible thermodynamic processes. The acoustic waves incident on the building element cause oscillations of the air in the closed or open pores of the light concrete structure. Owing to the viscosity of the air the oscillations in such pores are accompanied by friction and the kinetic energy of the oscillating air is transformed into heat energy. Moreover, under the action of the incident waves the particles of the light aggregate -- bound together by the cement stone -- oscillate, and owing to internal friction their kinetic oscillation energy is transformed into heat. /432

The open pores of light concrete are narrow channels of any shape that permit an air current to pass through them. The incident acoustic waves cause the air in the pores to oscillate; these oscillations are accompanied by friction and the air's kinetic energy is partially transformed into thermal energy. The energy is also transformed because of irreversible processes connected with the thermal conductivity of air and skeleton. When the skeleton -- light aggregate and cement stone -- begins to oscillate

additional thermal energy is generated because of internal friction in the skeleton's material, with a different modulus of elasticity.

Light concrete elements have a compound structure:

-- the skeleton of the structure's mass $m' = m'_1 + m_2$, i.e., ρ_a for 28 days;

-- the volume of air due to the structure's velocity p_{av} (the apparent porosity in volumes, determined in the laboratory).

For every brand of concrete used in the light aggregate the characteristics of interest were determined in the laboratory (Table 2).

TABLE 2

Chacteristic	Type of Concrete	Brand of concrete							
		B50	B75	B100	B150	B200	B250	B300	B400
Apparent density for 28 days $\left(\frac{kg}{m^3}\right)$	GC	1430	1660	1720	1750	1770	1800	1850	1900
	SC	1585	1780	1880	1950	1970	-	-	-
Modulus of elasticity $E\ \left(\frac{kg}{cm^2}\right).10^3$	GC	52	64	71	120	150	160	180	200
	SC	67,5	86	115	151,2	171,5	-	-	-
Apparent porosity P_{av} %	GC	20	18	16	11	7	6	5	5
	SC	18	14	12	9	7	-	-	-

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Porosity was determined in the laboratory for mechanically compacted samples (similar to the execution technology).

$P_{av} = (G_2 - G_1)/V_a$ is the apparent porosity in volumes;

G_2 is the weight of the sample saturated with water (kg/m^3);

G_1 is the weight of the sample in the dry state (kg/m^3);

V_a is the sample's apparent volume (m^3).

The greater the volume of open pores -- in general, the more inferior the brand of light concrete under constant compacting technology conditions -- the lower the sound damping coefficient. Thus, into Eq. (1) I have introduced the coefficient $i_p = 1 - p_{av}$, i.e., the coefficient of diminution of the sound insulation of the element's mass under the effect of the open pores.

Closing the pores on both sides of the building element by plastering, painting, etc., completely cancels the negative influence of the great volume of open pores on the sound reduction of the separating element. It is recommended to apply sound-absorbing paint. /433

An interesting sound reduction effect appears in the case of pores open only on one side of the building element's surface and closed on the other. In the resonance frequency range of the oscillations of the air contained in the material's pores intensive absorption of the acoustic energy can be observed, which leads to a considerable increase in sound reduction at these frequencies. It is recommended that the surface with open pores be oriented toward the noise source, if the position of the source is defined.

The incident wave will cause the skeleton to vibrate at one of its proper freqeuncies, depending on whether m'_1 or m'_2 predominates. Owing to the great amplitudes of the oscillations that occur at resonance (when the frequency of the incident wave coincides with or is very close to one of the proper frequencies), the transmission of noise through the structure will be greater than at other frequencies. In order to avoid the annoying effect of these resonances the structure's proper frequencies must be as low as possible. From this point of view the critical frequency f_c of a light concrete structure must be situated outside the main audio frequency range: 100 Hz $\geqslant f_c$ or $f_c > 3{,}200$ Hz.

The building element's thickness d for aerial noise insulation is imposed by constructional considerations (for example, supporting wall, resistance plate, etc.) or is determined with the aid of Eq. (1) from the law of mass. /434

It is proposed that the lowest coincidence frequency (critical frequency) be determined as a function of the characteristics of the component elements of the structure's skeleton:

$$f_c = \frac{c^2}{2\pi}\sqrt{\frac{12 \cdot m \cdot k_2}{d^3 \cdot E_{bu}}};[\text{Hz}]$$

where:

d is the thickness of the building element;

c is the velocity of sound in air;

E_{bu} is the longitudinal modulus of elasticity for the type and brand of light concrete;

m is the superficial mass of the building element ($m = \gamma \cdot d$)

$k_2 = m'_2/m'_1$ is the distribution coefficient of the cement stone (m'_2, sand + cement + physically and chemically bound water) referred to the mass m'_1 of the light aggregate, in the skeleton of the concrete.

The closer coefficient k_2 is to unity, the higher the critical frequency of the light concrete; for superior brands (B400) it behaves similarly to the coefficient for heavy concrete.

2. Impact Noise Insulation

Impact noise insulation is studied for floors.

The theory of the sound reduction of floors for impact noises is based on the acoustic power radiated by a plate with one or more layers, which is excited by periodic pulses that are successive in time.

Of the category of light concretes for floors, concretes of brands B200-B400 are used.

As in the case of heavy concrete, the most important factor in impact noise reduction is the thickness of the floor.

For plate thicknesses greater than 12 cm ($h_p \geq 12$ cm) impact noise absorption corrected in Eq. (2) by coefficient k_2, is greater /435 in light concrete plates than in heavy concrete plates.

Between theoretical results L_{Ncalc} and experimental measurements L_{Nexper} the differences are no greater than 2-4 dB.

At high frequencies ($f \geq 2{,}000$ Hz) considerable deviations occur, which can be explained by the lower rigidity of light concrete plates as compared with heavy concrete plates, for the same general technological characteristics.

Impact noise insulation must be studied as a function of the different types of flooring that go to make up the structure of the floor.

Conclusions

The use of light concretes in civil and industrial buildings is a recent development in Rumania. In order to ensure a broad field of application and a technological and economical efficiency that surpasses traditional solutions, intensive studies and applied research are necessary.

The results of the research conducted in Rumania up to the present time are in agreement with those obtained abroad, which constitutes a confirmation of our work.

In view of the many qualities of light resistance concretes (B150-B400) as compared with heavy concrete, their extensive utilization for walls and floors can be expected, especially in civil buildings.

Using light concretes leads to better acoustic comfort than do traditional solutions.

Summary

The author presents theoretical considerations about applied research on the sound-insulating capacity of building elements made up of light concretes. /436

Analyzing differentially the behavior of light concrete building elements under the influence of incident acoustic energy and on the basis of experimental measurements, the author introduces coefficients of correction into the basic formulas for calculating the sound-insulating capacity for the 100-3,200 Hz frequency band.

Studies and applied research will increase with the increasing utilization of light concretes in buildings with different functions.

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OPTIMIZATION OF THE ACOUSTIC ABSORPTION COEFFICIENTS OF CERTAIN FUNCTIONAL ABSORBENTS

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Today, mineral fiber plates and cushions are finding a broad /437
field of applications in the preparation of different types of sound-absorbent structures. The absorption of the acoustic oscillations that are incident on these materials is due to the friction losses of the kinetic energy of oscillation of the air in the pores of the material.

From the constructional point of view these sound-absorbing structures occur in two principal versions.

-- a porous layer of finite thickness on a rigid reflecting surface (wall, ceiling);

-- present between porous layer and reflecting surface is a certain air space.

In the first version, the frequency characteristic of the absorption coefficient shows a considerable drop in the low frequency region, a drop at a frequency that is the higher, the less the thickness of the porous material. It makes no sense to increase this thickness beyond certain limits because this leads to a deterioration of the adaptation of the wall's acoustic impedance to the transmission of sound energy from the air toward the structure, especially at low frequencies. As a matter of fact, close to the wall's surface friction losses become negligible anyway because of the flow velocity of motion of the air. In this way we arrive at the second type of sound-absorbing structure with a porous /438
layer which, however, has the disadvantage of maximum absorption only for certain frequencies.

These considerations led us to study the absorption coefficients of some sound-absorbing structures in which the porous material (FA/S-20 and AP/S-40 mineral wadding plates) were separately placed at an angle of inclination r with respect to the treated surface that varied from 0^{o} to 45^{o}.

In addition to the thickness d, the area of the plate surface S_a and the angle r, we also took into consideration the covering index s, defined as the ratio of the plate's designed area S_b (on

ceiling or wall) and the total area S_1 of the treated surface belonging to a plate (Fig. 1), i.e.:

$$a = \frac{S_b}{S_1} = \frac{S_a \cdot \cos r}{S_1} \qquad (1)$$

with values ranging from 0.4 (0.5) to 1.0.

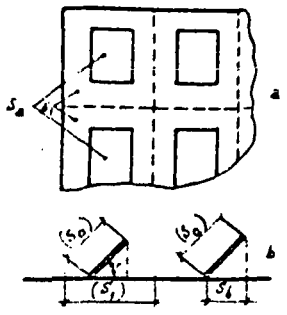

Fig. 1. Manner of placing sound-absorbing plates on supporting surface as a function of covering index (a) and angle of inclination.

The absorption coefficients were determined in samples with a surface ranging from 4 to 10 m^2 by the reverberation chamber method. The useful volume of the chamber was 120 m^3 and reverberation time for the empty chamber had the values shown in Table 1.

Analysis of the experimental results showed that if the parameter is angle r, the frequency characteristics of the absorption coefficients for a certain value of the covering index s generally keep this character, including a shift of the absorption maximum toward lower frequencies for plates with greater thicknesses. /439

It will be noted that maximum absorption occurs for values of angle r ranging from 15° to 30° (Fig. 2).

This behavior can be observed even more clearly if the absorption coefficients for the frequency bands of one octave are represented directly as a function of angle r, with s as the parameter (Fig. 3). On the other hand, representing the same coefficients as a function of covering indices s with r as the parameter does not reveal anything special, the variation of the absorption coefficient being of very little importance.

TABLE 1: REVERBERATION TIME T OF REVERBERATION CHAMBER

f(H_z)	125	250	500	1000	2000	4000	global
T(sec)	8,15	7,7	7,0	6,0	5,1	3,3	7,35

In view of the marginal effect for functional absorbents with sizes on the order of acoustic wave lengths, this latter result was to be expected. /440

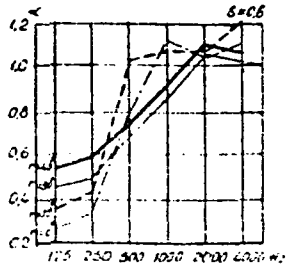

Fig. 2. Frequency characteristic of sound absorption coefficient for a certain value of covering index s as a function of angle of inclination r.

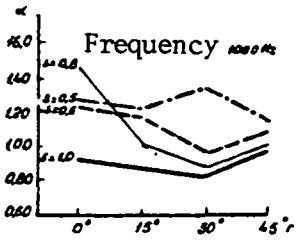

Fig. 3. Variation of the absorption coefficient for a frequency band of an octave as a function of angle r and with s as the parameter.

Under certain rather general conditions, such as for example, in industrial plants, where the normals to even surfaces and interior noise sources show an almost linear frequency characteristic, it can be shown that the typical noise level damping ΔL can be calculated as a function of the ceiling's absorption coefficients alone, i.e.:

$$\Delta L = 10 \log \left(1 + \frac{1}{2} \frac{\alpha - \alpha_0}{\alpha_0} \right) \qquad (2)$$

where $\alpha_0 = 0.5$ is the mean absorption coefficient of the surface of the plant (ceiling) before treatment.

Knowing the variation of the cost index c as a function of the same parameters r, s and d, we were then able to correlate this with the typical damping calculated in the above described manner for the purpose of determining the technological and economical efficiency expressed in the indices $E = \frac{1}{c} \cdot \frac{(dB)}{100\ lei}$, as can also be seen from the diagrams of Fig. 4.

For both thicknesses under consideration it turned out that the pair of parameters s = 0.8 and r = 15 leads to the highest technological and economical efficiency. Thus, at a cost price that is approximately half that of the conventional, our solution ensures a reduction in the noise level by 6-8 dB and even much more. /441

We applied sound absorption treatment with the optimum parameters to a plant with noisy pumping machines of the chemical industry. The results are shown in Fig. 5, with reductions in the noise intensity level by more than 10 dB, in intermediate frequency bands.

Using this relatively simple setup and materials manufactured in Rumania, our solution ensures a higher technological and

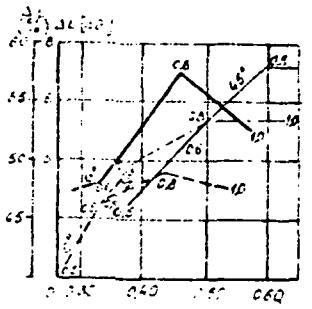

Fig. 4. Noise level damping curves ΔL as a function of cost index c.

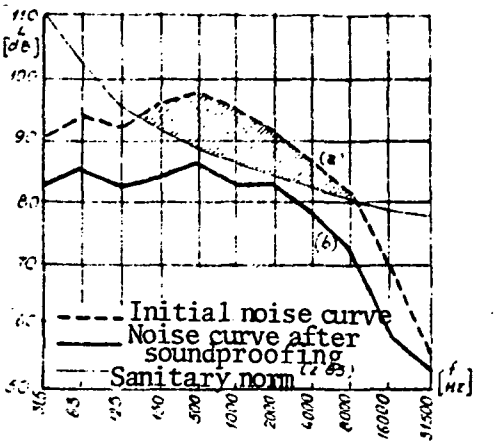

Fig. 5. Frequency characteristics of noise level in an industrial plant before (a) and after (b) sound absorption treatment.

economical efficiency than traditional solutions (for applying sound-absorbing materials). In view of the great number of independent parameters, the absorption coefficients were determined for a relatively small number of their values, so that of interest in the future would be research on a number of values of the parameters close to the optimum.

Summary

The author describes the main experimental results of research /442 on the sound absorption coefficients of some functional absorbents (mineral wool plates manufactured in Rumania), determined by the reverberation chamber method.

The influence of the angle of inclination of the sound-absorbing material with respect to the surface to be treated is analyzed for values of 0°, 15°, 30° and 45°, as well as the influence of the covering index, defined as the ratio of the designed area (on the surface to be treated) of a plate and the area of the treated surface belonging to a plate with values ranging from 0.4 to 1.0.

As compared with the conventional method of applying sound-absorbing plates, the analyzed structures have a higher technological and economical efficiency. The optimum structure corresponds to an angle of inclination of 15° and a covering index of 0.8.

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SOUND-ABSORBING SLABS AND STRUCTURES BASED ON GRANULAR MATERIALS (BOUND AND UNBOUND)

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In the present communication we describe a number of sound-absorbing slabs and structures based on granular materials (bound and unbound) and the methods of preparing these elements at the building site. /445

The above slabs (which are the object of patent number 71.862/4.08.1972) belong to the category of porous sound-absorbing structures used in the soundproofing of theaters and social-cultural and industrial buildings.

The best known slabs of this category are sound-absorbing slabs based on fibrous materials (mineral wool, glass wool, porous PFL, etc.) whose manufacture is possible only in specialized plants because it presupposes a complicated technological process, which leads to relatively high manufacturing costs. Moreover, these materials have low mechanical strengths, so that during transport toward the building site a significant percentage of the delivered quantity deteriorates. To this same category belong certain foams from plastics (polyurethane), but they are not much used in Rumania and the problem of the fire protection of these products has not yet been solved. Also known are wet and dry sound-absorbing plasters in which open pores are obtained by adding gas-eliminating products to the fresh mortar. But these plasters have a low absorption coefficient and require the use of relatively expensive additives.

The sound-absorbing slabs and structures that are the object of the present communication consist in principle of bound (in the case of slabs) or encased (in the case of structures) single-grained powders (sand, expanded blast furnace slag, etc.) that ensure finished products with an apparent porosity of 25-45% and a dissipation of energy in the structure leading to absorption coefficients comparable with those of mineral wool and polyurethane products.

The slab manufacturing process consists of three phases that can be carried out under building site conditions:

-- sorting of the granular material;

-- preparation of the casting mixture;

-- casting the mixture into slabs.

The granular material is sorted for the purpose of obtaining a single-grained fraction.

The mixture is prepared by adding to the sorted material a binder (cement + water, epoxid resin + hardener, etc.) in a suitably chosen proportion so as to modify as little as possible the porosity of the powder in bulk and at the same time ensure the necessary slab strength after hardening. /446

The mixture is cast in ordinary casings and the slabs can be struck and assembled after hardening.

In what follows we shall give two examples of sound-absorbing slab preparation:

Example 1:

Sound-absorbing slabs based on sand and cement. A mixture with the following weight composition is prepared: 8 parts sand with a granulation of 0.5 to 0.8 mm, 2 parts Pz 400 cement, 1 part water.

This mixture is cast in brand M 100 mortar frames with which they form a common body.

The outer dimensions of the frame can be, for example, 30 x 30 x 2 cm and the inner dimensions: 25 x 25 x 2 cm.

These slabs are characterized by the following sound absorption coefficients α, determined by the interferometer tube method:

f (Hz)	100	200	400	800	1000	1250	1600	2000	2500	3150	4000	5000	6300
	0,04	0,06	0,12	0,29	0,48	0,70	0,76	0,80	0,65	0,60	0,60	0,60	0,82

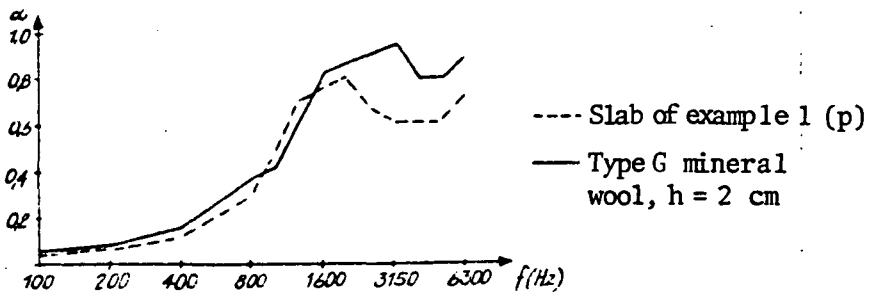


Fig. 1.

Example 2:

Sound-absorbing slabs based on sand and epoxid resins. A mixture with the following weight composition is prepared: 150 parts sand with a granulation of 1 to 1.25 mm, 10 parts Araldit F, one part hardener for the Araldit F, 2.5 parts diluent for the Araldit F. This mixture is cast in casings of any type with a view to obtaining slabs that can have dimensions of, for example, 25 x 25 x 2 cm.

These slabs are characterized by the following sound absorption coefficients, determined by the interferometer tube method: /447

f (Hz)	100	200	400	800	1000	1250	1600	2000	2500	3150	4000	5000	6300
	0,07	0,18	0,17	0,32	0,36	0,50	0,63	0,96	0,90	0,70	0,50	0,54	0,54

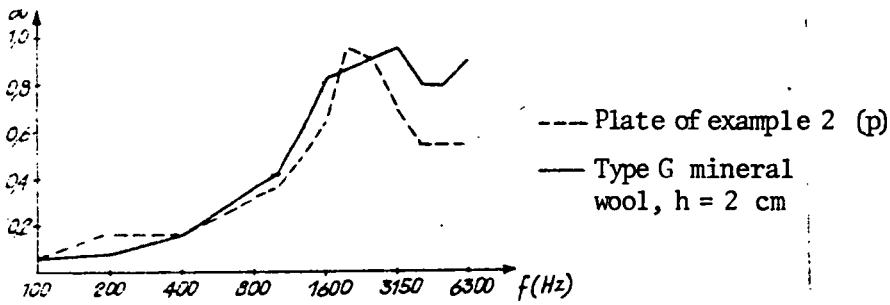


Fig. 2.

These slabs can be used either in the rough or painted with water colors, which do not change their sound-absorbing properties.

The above described compositions are also applied as the last layer, the seen side, on prefabricated elements of reinforced concrete. Let us mention that such a solution was included in the project for soundproofing the auditorium of the Trade Union House of Culture in Arad, which is to be built in the course of the years 1973-1974.

In addition to the above-mentioned slabs, the authors also studied a number of structures made of granular, unbound, encased material whose principal advantage is a substantial increase in the absorption coefficients α in the low-frequency region.

The procedure of manufacturing the structures consists of four phases that can be implemented at the building site:

-- sorting of the granular material;

-- preparation of the cases;

-- filling of the cases with the sorted material;

-- closing of the cases.

As in the case of the slabs, the granular material is sorted with a view to obtaining a single-grained fraction.

The cases are made of sorted material mixed with binder (by casting) or, possibly, of other materials (wood, metal) in the shape of a prism and the side that is to be seen is kept open in order to be able to fill the interior with the chosen powder.

The cases are closed by applying to the frame some materials that do not change the sound-absorbing qualities of the products (unbitumenized glass fiber felting covered with a perforated foil of vinyl polychloride, aluminum, etc.). /448

In what follows we shall give two examples of preparing sound-absorbing structures:

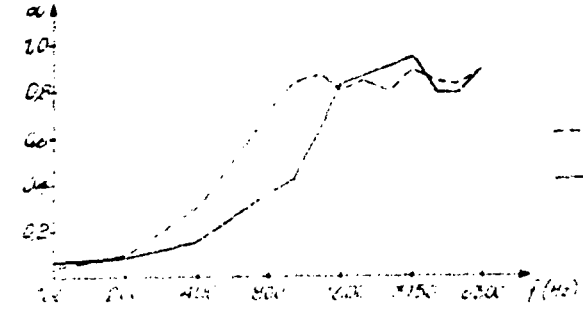
Example 1 (s):

Sound-absorbing structure based on crushed blast furnace slag, prepared in a case of sand bound with cement. The structure is made up of a case formed of a brand M 100 mortar frame 6 having, for example, outer dimensions of 30 x 30 x 4 cm and inner dimensions of 25 x 25 x 4 cm and a background slab 7, with a thickness of, for example, 2 cm, cast inside the frame of a mixture prepared as in example 1.

After hardening, the case is filled with blast furnace slag 3 and is closed with glass fiber felting 4 glued to the frame with an adhesive based on vinyl polychloride on top of which a rigid perforated foil of vinyl polychloride 5 is applied by gluing with the same adhesive.

This structure is characterized by the following sound absorption coefficients α, determined by the interferometer tube method.

f (Hz)	100	200	400	800	1000	1250	1600	2000	2500	3150	4000	5000	6300
α	0,05	0,10	0,32	0,72	0,85	0,87	0,90	0,85	0,80	0,89	0,85	0,85	0,90



---Slab of example 1 (s)

——Type G mineral wool, h = cm

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Fig. 3.

Example 2 (s):

Sound-absorbing structure based on crushed blast furnace slag, prepared in a case of sand bound with epoxid resin. The structure consists of a case 8 having, for example, outer dimensions of 30 x 30 x 4 cm and inner dimensions of 25 x 25 x 2 cm and entirely made up of a mixture prepared as in example 2, filled after hardening with blast furnace slag 3. The case is closed with glass fiber felting 4, coated with epoxid resin on top of which a rigid perforated foil of vinyl polychloride 5 is applied by gluing. /449

This structure is characterized by the following sound absorption coefficients α, determined by the interferometer tube method (see Fig. 4).

Let us mention that if an ordinary material (wood, metal) is used in manufacturing the case, we no longer get such good values for it as in the preceeding cases, in the low-frequency region.

We have estimated that the cost of soundproofing with the above slabs and structures is approximately as follows:

1) sound-absorbing slabs based on sand and cement...45-50 lei/mp;

2) sound-absorbing slabs based on sand and epoxid resin... 110-120 lei/mp;

3) sound-absorbing structure based on crushed blast furnace slag, prepared in a wooden case ..... 60-70 lei/mp;

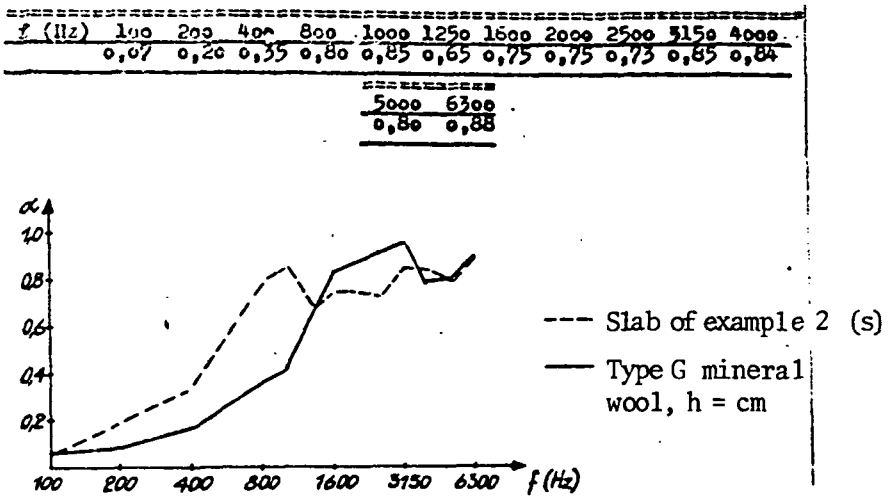

Fig. 4.

4) sound-absorbing structure based on crushed blast furnace slag, prepared in a case of sand bound with cement ... 80-85 lei/mp;

5) sound-absorbing structure based on crushed blast furnace slag, prepared in a case of sand bound with epoxid resin .... 135-145 lei/mp.

As compared with the sound-absorbing materials used at the present time in the Rumanian Socialist Republic the following should be mentioned:

-- Sound-absorbing slabs based on sand and cement are cheaper /450 than the FA 440-S slabs now in use and yield similar results;

-- The above described structures can be manufactured more economically than existing structures with the same acoustic performance;

-- For the time being, sound-absorbing slabs based on sand and epoxid resin cannot be regarded as a practical solution inasmuch as the binder is still too expensive in Rumania.

It should also be mentioned that all the above described slabs and structures can be manufactured under industrial conditions as well as under building site conditions inasmuch as the basic materials are easy to obtain and the manufacturing processes are relatively simple.

Summary

The authors describe sound-absorbing slabs and structures made up of bound or unbound granular materials and how to manufacture these elements at the building site. The raw material is a single-grain powder (sand, expanded blast furnace slag, etc.) that imparts to the end products an apparent porosity of 25-45% and an energy dissipation within the structure leading to absorption coefficients that can be compared with those of mineral wool and urethane.

/451

N74-28130

COMPLEX SOUNDPROOFING OF INDUSTRIAL ROOMS

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One of the noxious factors that are polluting the environment /453
as a direct consequence of the development of industry and means of conveyance, is urban noise. Owing to the harmful effects of noise on the human organism as well as its negative influence on productivity, noise control in residences, social-cultural buildings and industrial plants is an urgent necessity. If the noise source can not be insulated or insulation at the source cannot be applied for technological reasons, noise in enclosed spaces is controlled with the aid of soundproofing.

The choice of sound-absorbing material depends, in the first place, on the spectrum of the emitted noise, and the material's maximum efficiency must be in the frequency region that corresponds to the highest noise level. The efficiency of different sound-absorbing structures depends on their internal mechanism of absorption, i.e., the structure and nature of the materials employed, the thickness and distance of these from the room's surfaces, etc., as a function of the frequency region of interest without, however, neglecting the economic aspect.

In what follows we shall describe some sound-absorbing structures manufactured or about to be manufactured for application in /454
industrial rooms with a high noise level, i.e., a level that exceeds the admissible noise level (the C_z85 curve) by about 7-20 dB for different frequency regions.

Thus, in the case of an industrial room equipped with Rutti looms, the noise level exceeds the C_z85 curve by 7-9 dB for the 800-7,000 Hz frequency range (Fig. 1, curve 1).

Soundproofing the ceiling (which accounts for 1/3 of the room's interior surface) by means of 60 x 120 cm panels led to a reduction in the noise level by 4-7 dB (Fig. 1, curve 2). The panels were made of 5 cm thick mineral wool cushions supported on a metal frame on which a wire net was mounted. For protection, sackcloth was inserted between cushion and wire net (Fig. 2). To be noted is the fact that owing to objective conditions the panels were applied 75 cm from the floor board. Maximum values of the absorption coef- /455
ficients in the 700-8,000 Hz frequency range (measured with an acoustic interferometer) were shifted to the 250-700 Hz frequency range (Fig. 3).

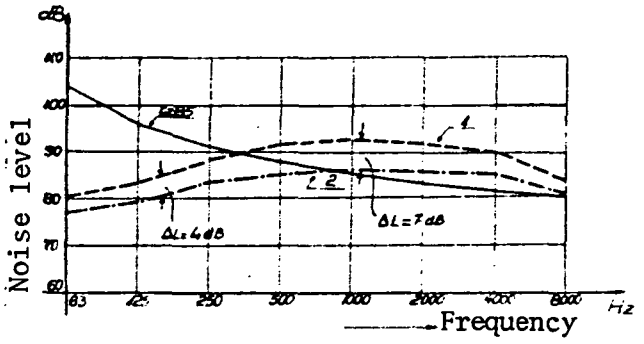

Fig. 1. Noise spectrum in industrial room. 1. Before treatment; 2. After treatment.

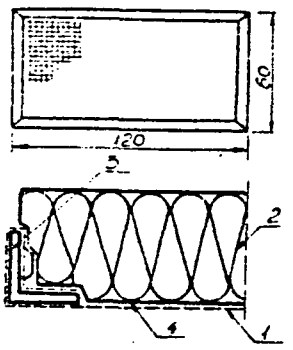

Fig. 2. Sound-absorbing panel. 1. Rustproof sieve; 2. 5 cm thick mineral wool cushion; 3. 20 x 20 x 31 angle-iron profile; 4. Sackcloth.

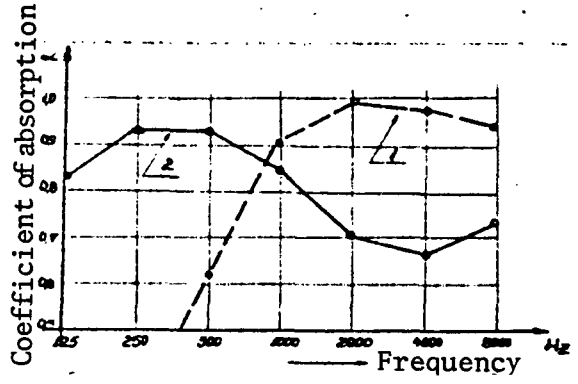

Fig. 3. Variation of absorption coefficient with frequency. 1 - absorption coefficient values determined with acoustic interferometer 2 - absorption coefficient values measured after treatment.

The minimum value of the absorption coefficient α (0.66) is located around the 4,000 Hz frequency region, where we ought to have an absorption maximum. As a consequence, as we expected, the air space has an unfavorable effect for the frequency band of interest. Should the panels be applied at a lesser distance from /456 the ceiling, with an insignificant increase in the surface treated, the noise level can be reduced below the C_z85 curve.

We recommended soundproofing the ceiling and walls of a thermopower plant equipped with gamma boilers, where the C_z85 curve was exceeded by 8-14 dB for a wide frequency region. For reasons similar to the preceeding, the absorbing ceiling, which was made of FA/140/S slabs, was applied at a distance of about 50 cm from the roof slab. The walls were lined with mineral wool cushions protected on the outside by sackcloth and wire net (Fig. 4).

In order to prevent mechanical deteriorations we made provi- /457 sion at a height of 1.5 m for protective masonry of bricks with their holes oriented toward the room (Fig. 5).

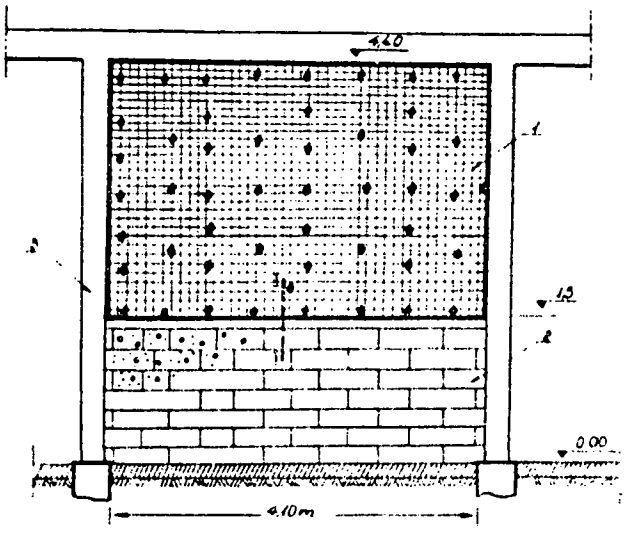

Fig. 4. Soundproofing in a current bay. 1. Mineral wool cushion protected by wire net; 2. Protective masonry; 3. Reinforced concrete post.

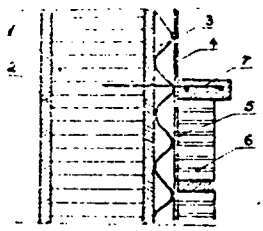

Fig. 5. Section of soundproofed wall. 1. Masonry wall; 2. Plastering; 3. Mineral wool cushion; 4. Sackcloth; 5. Rustproof wire net; 6. Protective layer of brick with holes; 7. Reinforced concrete girdle.

Approximative estimation of the reduction in the noise level ΔL was determined as a function of frequency by using the reverberation times calculated with and without soundproofing (Table 1). The real absorption characteristics must be determined after the wall and ceiling works have been terminated.

TABLE 1: NOISE DAMPING BY SOUNDPROOFING OF WALLS AND CEILING

ΔL (dB)	Frequency in Hz					
	125	250	500	1000	2000	4000
Relative decrease in noise level in dB	13	14.5	15	16	14.5	14

In a room that houses machines for recuperating solid wastes from plastics, the noise level exceeds the admissible level by 10-20 dB for the frequency bands ranging from 500 to 8,000 Hz (Fig. 6).

In this case we adopted a complex treatment with a maximum absorption surface. In addition to treating walls and ceilings for sound absorption, we opted for 60 x 200 cm suspended panels made of cushions of felt-protected mineral wool and specially oriented and /458 with different angles of inclination with respect to the source.

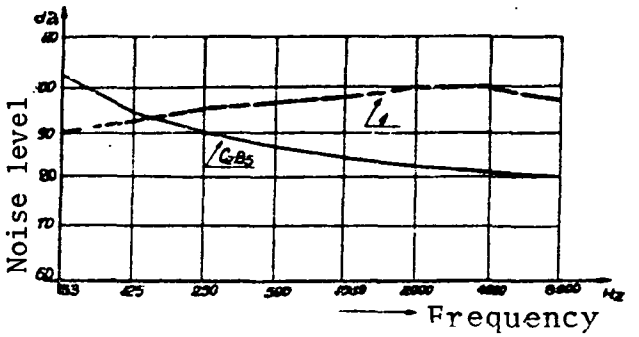

Fig. 6. Noise spectrum of waste recuperating room as compared with C_z85 curve. 1. Noise level measured in dB.

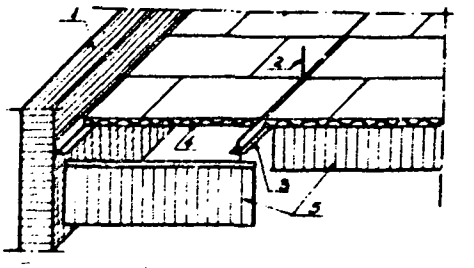

Fig. 7. Treatment applied to waste recuperating room. 1. Brick wall; 2. Hanging moustache; 3. Plate supporting skeleton; 4. Sound-absorbing plates; 5. Suspended absorbing panels.

The panels are suspended from the resistance skeleton of the sound-absorbing ceiling (Fig. 7).

Without taking into account the influence of the suspended panels we used the reverberation times calculated with and without soundproofing to determine the relative reduction in the noise level /459 for the 125-4,000 Hz frequency range (Table 2), for which we obtained minimum values of 12.5 dB in the low-frequency region and maximum values of 17 dB in the intermediate-frequency region.

TABLE 2: NOISE ATTENUATION IN THE CASE OF COMPLEX TREATMENT

ΔL (dB)	Frequency in Hz					
	125	250	500	1000	2000	4000
Relative decrease in noise level in dB	12,5	13,5	16	17	17	16,5

Future measurements will permit determination of the real efficiency of these treatments taking into account their order of execution.

By comparative studies in rooms of the same size it will be possible to estimate quantitatively the real efficiency of combining the soundproofing of walls, ceilings and suspended structures.

Summary

The authors describe some structures treated for sound absorption that are used to soundproof industrial rooms with a very high noise level (weaving sheds, rooms for the recuperation of solid wastes, thermal power plants, etc.) using materials manufactured in the Rumanian Socialist Republic. /460

Depending on the dimensions in plan of the rooms, the authors adopted soundproofing treatments for the walls and ceilings or only for the ceilings.

In the case of relatively small rooms having a noise source with a high level, the authors recommended complex treatments involving, in addition to soundproofing of the walls and ceiling, suspended panels specially oriented with respect to the noise source. The efficiency of the adopted solutions was analyzed experimentally after their execution and compared with the calculated damping values.

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N74-28131

HIGHLY EFFICIENT SPECIAL SOUND-ABSORBING SOLUTIONS

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In the period 1970-1973, INCERC undertook a number of studies /461 for the purpose of producing sound-absorbing structures on the basis of materials manufactured in the Rumanian Socialist Republic.

As a first step we studied solutions involving plywood.

Our results were published in an article entitled "Sound-absorbing structures made of new materials manufactured in the RSR" published in No. 6/1971 of the magazine "Constructii" (authors: Serban Petre-Lazar and Mariana Ionescu) and on the basis of these results we worked out "technical instructions for using sound-absorbing structures based on type G mineral wool slabs" (C 146-72).

Our subsequent studies centered around the problem of producing highly efficient special structures with the following criteria in mind:

A) Bringing about a distribution surface of the sound-absorbing material greater than that of the building element on which the structure is placed, and obtaining a large absorption surface on a limited geometrical surface.

This criteria was adopted in view of the fact that in very many cases, especially in industrial plants, sound-absorbing systems cannot be arranged except on the ceiling of the room (sometimes even there only partially).

Under these conditions the absorption surface that is obtained by using prefabricated FA slabs or other solutions is sometimes insufficient, so that structures must be used which offer sound waves a greater surface than the geometrical surface supporting them.

B) Obtaining the highest possible absorption coefficient in the widest possible frequency band.

C) Creation of light elements that can be adapted to different constructional and aesthetic conditions.

Of the different types of structures studied, we shall discuss a covering of vertically arranged mineral wool slabs.

The slab covering can be mounted on a metal skeleton, as can be seen from Fig. 1.

-PERSPECTIVE-

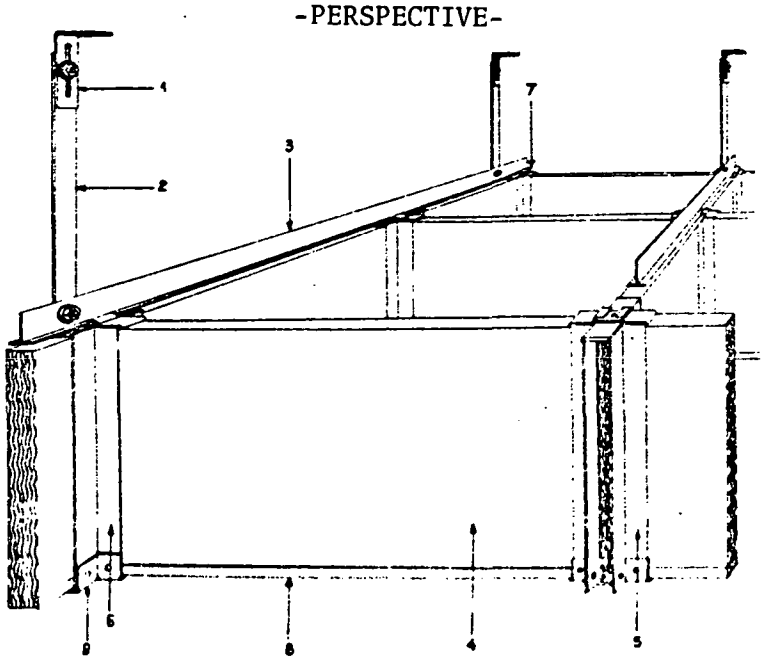


Fig. 1. Covering of vertically arranged mineral wool slabs. 1. Gripping angles; 2. Distance piece; 3. Inverted T supporting section; 4. Mineral wool slabs; 5. Supporting element (current field); 6. Marginal supporting element; 7. Corner element; 8. Supporting rail of wool slab; 9. Closing collar.

/463

In the case of a horizontal supporting surface, sound-absorbing slabs or any other sound-absorbing structure can be mounted on this slab by gluing or some other procedure, with the slab covering being supported from below, as indicated.

The covering of vertically arranged mineral wool slabs can be applied to buildings of any degree of fireproofing.

In order to guard the slabs against deteriorations, as well as in order to give them a uniform appearance, it is recommended that they be covered with protective felt on both sides.

This operation can be implemented by applying the filter in the normal manner to a single side of a 2 cm thick slab and then gluing two slabs together by their unlined sides.

The sound-absorbing qualities were determined by the reverberation chamber method.

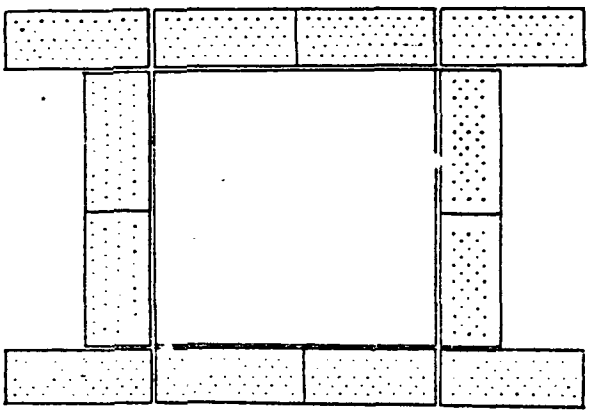
In order to elucidate the manner in which the vertically arranged sound-absorbing slabs work we carried out determinations on the three structures shown in Figs. 2-3.

The results obtained are shown in Table 1 and in Figures 4-6.

/466

Analysis of the above results leads to the following conclusions:

1. In the case of structures II and III, for a 20 mm thick covering of type G mineral wool slabs, we obtained absorption



-plan-

Fig. 2. Structure I. (All slabs are vertically arranged)

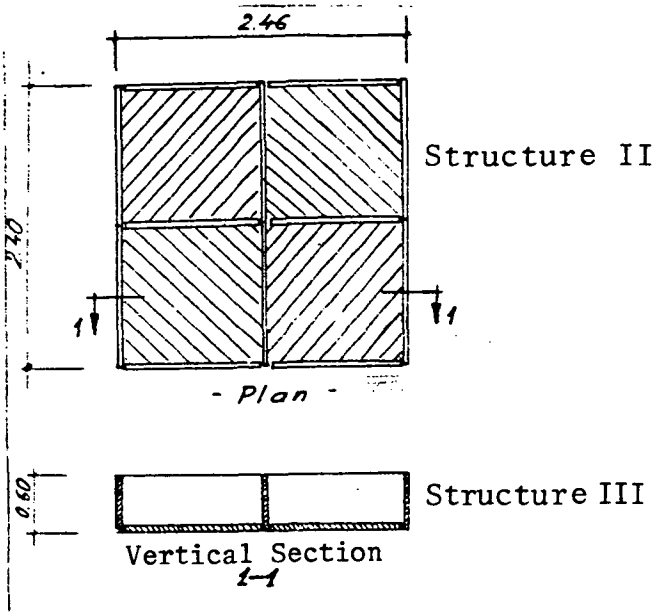


Fig. 3. Structure II. (Covering of vertically arranged type G mineral wool slabs (h = 20 mm) plus type G mineral wool slabs arranged horizontally in eyes of the covering. Note: Structure III has no type G mineral wool plates in the eyes of the covering.

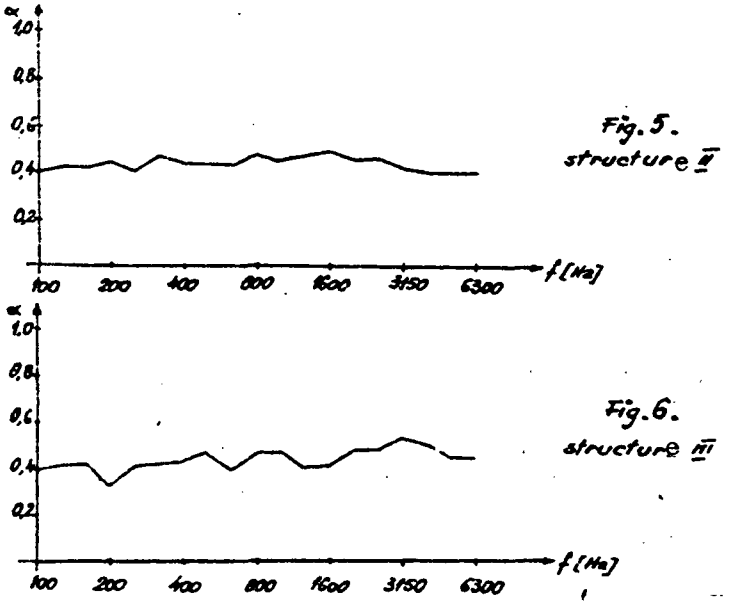
TABLE 1: COEFFICIENTS OF ABSORPTION α OF THE INVESTIGATED STRUCTURES, CALCULATED WITH DUE REGARD FOR THE SURFACES OF BOTH SIDES OF THE SLABS

f(Hz) / STRUCTURE	100	125	160	200	250	315	400	500	630	800	1000
I	0,07	0,11	0,13	0,20	0,26	0,33	0,43	0,70	0,70	0,78	0,81
II	0,40	0,42	0,42	0,44	0,40	0,47	0,44	0,44	0,43	0,48	0,45
III	0,40	0,42	0,42	0,43	0,41	0,42	0,43	0,47	0,40	0,47	0,47

1250	1600	2000	2500	3150	4000	5000	6300
1,04	1,04	0,97	0,97	0,90	0,90	0,85	0,85
0,47	0,48	0,46	0,46	0,42	0,40	0,40	0,40
0,40	0,41	0,48	0,48	0,53	0,50	0,45	0,45

coefficients with a value of 0.4-0.5 in the entire useful frequency range (100-6,300 Hz), if we considered that this coefficient corresponded to a square meter of the slab covering's geometrical surface calculated with due regard for both sides of the slab (or 0.8-1.0,

Fig. 4. Note: In calculating the absorption coefficient for structures II and III, we used the geometrical surfaces resulting from consideration of a single side of the vertically arranged slabs.



Figs. 5 and 6. Note: In calculating the absorption coefficient for structures II and III, we used the geometrical surfaces resulting from consideration of a single side of the vertically arranged slabs.

f, we have still another explanation of the a covering of vertical slabs the absorption with the frequency.

if we considered a single given surface of the slabs).

What must be stressed, in particular, is the fact that the absorption surfaces of the slab covering are approximately constant.

In our opinion, this can be explained by the fact that in the present case it is not only a question of a transformation of the oscillation energy of the volume of air into thermal energy (by friction) but also a transformation into oscillation energy of the vertically arranged slabs under the influence of sound waves as well as certain dynamic loads. Inasmuch as the slabs are continuous bodies, they are characterized by an infinity of proper frequencies, which for $f > 100$ Hz are thence enough for no special resonance zones to exist (for which the absorption coefficients would have to have higher values). Taking into account the fact that the oscillation energy of a body is a function of the product $(Af)^2$ (where A is the oscillation amplitude and f is the oscillation frequency) in which A decreases with increasing fact that in the case of coefficient does not vary

2. Comparing the absorption surfaces that were obtained in the case of the structures I (horizontally arranged slabs) and II (covering of vertical slabs plus slabs arranged horizontally in the eyes of the covering) we found that in the 100-630 Hz frequency band the absorption surface is greater in the case of structure II, and for frequencies above 630 Hz the absorption surfaces are approximately equal. In view of the fact that structure II is placed on a geometrical surface of 5.80 m^2 and structure I, on a surface of 14.40 m^2, the results prove that in the case of the covering of vertically arranged slabs a great absorption surface was obtained on limited geometrical surfaces, as we proposed to do on the basis of the criterion enunciated in the beginning.

Our results were confirmed in practice by applying solution III to the acoustic treatment of a polyvalent room with 350 seats that was designed by ISART.

The research of our group on special sound-absorbent solutions is continuing at the present time with the aim of determining the dependence between the magnitude of the angle between the sound-absorbing surface and the supporting surface and the sound-absorbent qualities of the solutions.

Summary /467

The authors present some results of INCERC research aimed at bringing about highly efficient special sound-absorbing structures with the following criteria in mind:

-- Bringing about a distribution surface of the sound-absorbing material greater than that of the building element on which the structure is placed, i.e., obtaining a great absorption surface on limited geometrical surfaces.

-- Obtaining the highest possible absorption coefficient in the widest possible frequency band.

-- Creating elements adaptable to different constructional and aesthetic conditions.

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SOUND ISOLATION PROPERTIES OF A LIGHTWEIGHT DWELLING HOUSE.

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SOUND ISOLATION PROPERTIES OF A LIGHTWEIGHT DWELLING HOUSE

Summary

The opinion about prefabricated buildings to the effect that they are inferior to traditional ones,seems to be proved by difficulties with sound isolation between dwellings in the most systems of prefabrication. The use of elements of extremely low mass and a simple structure makes the meeting of sound isolation demands nearly impossible. Regarding that,various lightweight elements in diferent arrangements were designed and taken into consideration to prove or to reject the convelence of application of them in dwelling houses from the point of view of building acoustics. From values of mass calculated as values necessary for arrangements considered in order to reach the value of sound isolation demanded , the lowest one was choosen for the realization of a dwelling house. Though the mass of elements as walls and floors do not exceed 80 kg/m^2 the results reached by measuring the sound isolation between dwellings well answers the values preliminary demanded .Both the details and sound isolation properties of the final form of building are reproduced.

/469